MARINE PROPELL

SYDNEY W. BARNABY

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PROPELLERS

BY

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M. INST. C.E. AND M.I.N.A.

FOURTH EDITION

London

E. & F. N. SPON, Ltd., 125 STRAND

New York

SPON & CHAMBERLAIN, 12 CORTLANDT STREET

1900

PREFACE

TO

THE FOURTH EDITION.

SINCE the publication of the third edition of this book the phenomenon of "Cavitation" has engaged much attention. It has already considerably affected the design and arrangement of screws for vessels of very high speed, and it is difficult to predict what its influence upon future progress will be. It seems most likely to be in the direction of a greater subdivision of power in order to reduce the diameter of screws and thus increase both their rate of revolution and their immersion.

In the chapter devoted to Cavitation will be found all that is yet known about the subject. I have compiled a table showing the cavitating speed of a propeller of normal form and moderate immersion at different pitch-ratios and slip-ratios, and I have given a formula by means of which the cavitating speed under any conditions can be calculated. The table shows at a glance the

relative effect of change of pitch-ratio and change of slip-ratio upon the speed at which cavitation commences.

The formula for the minimum area of blade necessary to avoid cavitation will, I think, be found useful by all those who have to design screws for high-speed vessels.

SYDNEY W. BARNABY.

THE HOLLIES, CHISWICK MALL, W.
January, 1900.

PREFACE

TO

THE THIRD EDITION.

In issuing a third edition of this book it appeared advisable to reconstruct and enlarge it in order to introduce new material which had become available. The original form of lectures as delivered to a class of students has been abandoned, and the book rewritten and brought up to date.

No attempt has been made to give a historical review of marine propulsion, and of the innumerable forms of screw proposed since 1836, only those are described which embody some intelligent idea.

The table of constants for disc-area and revolutions on page 108 will, I think, be appreciated by those who have many screws to design. The complete series of model experiments upon which it is based is the work of Mr. R. E. Froude, and I am indebted to him for permission to make use of it.

There are many ways in which it is possible to tabulate experimental results, but after much con-

sideration I believe that the table of constants which I have here given is the best that can be devised, being independent of scale and equally accurate for all sizes of propellers. It is possible by means of it to design a screw which shall have maximum efficiency under any given conditions of indicated horse-power and speed; or if revolutions or diameter are so limited as to preclude the adoption of the most suitable dimensions, those may be selected which will be the best under the given conditions and the efficiency at once ascertained, provided only that the vessel is of such a form that the propulsive coefficient is not abnormal, and that the designer can correctly estimate the speed of the following current in which the screw works. More than this I do not think can be reasonably expected. No table can supply the place of judgment and experience.

SYDNEY W. BARNABY.

THE HOLLIES, CHISWICK MALL, W.
September 14, 1891.

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MARINE PROPELLERS.

CHAPTER I.

FIRST PRINCIPLES.

THE principle upon which nearly all marine propellers work is the projection of a mass of water in a direction opposite to that of the required motion of the vessel.

If the weight of the mass of water acted upon by the propeller in pounds per second = W, and if the sternward velocity in feet per second imparted to it in relation to still water = S, then the reaction which constitutes the propelling force is $\frac{W\,S}{g}$, where $g = 32 \cdot 2$ feet per second; and this is independent of the form of propelling apparatus altogether. S is commonly known as the real slip, but will here be generally referred to as the rate of acceleration, or, more shortly, as the acceleration.

When the vessel is in motion at a regular speed, the reaction $\frac{W . S}{g}$ is equal to the resistance.

So long as there is a resistance to be overcome by the propeller, there is no possibility of reducing the real slip or acceleration S to zero, since a necessary condition would be that W, the weight of water acted upon, was infinitely large.

When a propeller is to be designed for any given set of conditions, it is of the first importance that the relation between the mass of water acted upon and the acceleration imparted to it should be such, that while the product $\frac{W\,S}{g}$ shall equal the estimated resistance of the ship, and the size and rate of motion of the propelling apparatus such as shall suit the conditions of the case, the economic result may yet be the best attainable, or may only fall short of the maximum by an amount which is calculable, and which it may be desirable to sacrifice in order to obtain other advantages.

The method of calculating the propulsive effect of the screw from the backward slip is that adopted by Rankine. Professor Greenhill has published some able papers on the theory of the screw,* in which he determines the propulsive effect as due to the reaction of the water to the rotatory motion given to it in the wake; and Professor Fitzgerald has shown,† that if the motion imparted to the water by a particular screw is assumed to be a certain form of vortex, and a calculation be made of the

* Trans. Inst. Naval Architects, xxix. p. 319.

† *Engineer*, Aug. 22 and Sept. 19, 1890.

power absorbed in producing such a vortex, it will be found to agree very well with the actual horse-power put through the screw.

As these independent investigations lead to the same conclusions as the older treatment, and as the latter is simpler and of more general application, it has been thought best to adhere to it.

Rankine has defined * the theoretical limit towards which the efficiency of propellers may be made to approximate by mechanical improvements, and has pointed out certain causes which make the actual efficiency fall short of that limit. He states that "if the propelling instrument be so constructed as to act upon each particle of water at first with a velocity equal to the velocity of feed"—that is the speed of the water entering the propeller —"and gradually increasing at a uniform rate up to the velocity of discharge, then the loss of work is the least possible." The oar, when a uniform force is applied to it by the oarsman throughout the stroke, approaches closely to this limit, as does also the screw-turbine propeller (see page 173).

There is a certain quantity of work which must be lost under all circumstances, and it is equal to the amount of energy of the discharged water moving astern with a velocity S relative to still water.

* *Engineer*, 1867, xxiii. p. 25; and 'Miscellaneous Scientific Papers,' p. 544.

As this energy varies as the weight multiplied by the square of the velocity, it follows that if the quantity of water acted upon is doubled, the loss from this cause is doubled, but if the acceleration is doubled, the loss is increased fourfold. This explains why the hydraulic propeller, which is forced to act upon a much less area of column than the screw, appears at such a disadvantage when compared with it.

The causes of loss of work in propellers of different kinds may be thus summed up:—

1. Acceleration of feed-water before its arrival at the propelling instrument. The radial paddle and the screw, and to a less extent the feathering wheel, are defective in this respect, the deficiency manifesting itself in the form of wasteful slip. The Ruthven pump, the oar and the screw-turbine propeller are more or less exempt.

2. Transverse motion impressed on the water. Propellers which lose in efficiency from this cause are ordinary screws which impart rotary motion; radial wheels, which give both downward and upward motion in entering and leaving the water; and oars, which impart outward and inward motion at the commencement and end of the stroke respectively. This loss is greatly reduced in the screw-turbine, and may be entirely avoided in the hydraulic propeller.

3. Waste of energy of the feed-water. This

is experienced by the hydraulic propeller only, as generally applied, and has been one of the causes of its inefficiency. It is not, however, a defect inherent to it, and has been avoided in some recent applications.

CHAPTER II.

THE PADDLE-WHEEL.

As a propelling instrument the paddle-wheel is not inferior to the screw, but its speed of revolution is necessarily slow, and paddle engines are therefore larger, costlier and heavier than screw engines of the same power. Until the introduction of the screw-turbine, it was the only propeller used for vessels of very shallow draught.

In order to ascertain the comparative value of the paddle and screw for towing purposes, H.M.S. *Rattler* and *Alecto*, the former a screw and the latter a paddle vessel of the same size and power, were lashed stern to stern. The *Rattler* towed the *Alecto* astern against the whole power of her engines at the rate of 2·8 knots. Almost as much power can be developed in a screw vessel when she is towing as when she is running free, but this is not the case in a paddle vessel. The engines of the *Alecto* could not get away, and were only able to develop 140 I.H.P., while the *Rattler* was developing 300.

It is difficult to frame rules for determining the proper area of floats for a given I.H.P. and speed of vessel, because so much depends upon

the position of the water surface in relation to the wheel when the vessel is at full speed.

There is frequently a wave hollow in way of the wheel due to the motion of the vessel, and the action of the paddles is to cause the water to run towards them, and to produce a still greater depression of level in front of, and below the wheel. Unless the vertical position is properly arranged the immersion of the floats will be insufficient at full speed, and the slip will be excessive.

The speed at which water can flow under the influence of gravity at a depth h below the surface is equal to $\sqrt{2gh}$. If the float of a paddle-wheel moves with a velocity v relative to still water, it is only at a depth h below the surface $= \frac{v^2}{2g}$ that the water will be able to keep up with the float, and all that part of the float which is immersed to a depth less than h will be denuded of water at the back. Thus, for example, in the case of a wheel having floats 3 feet 6 inches deep and having a slip—i.e. velocity of float in relation to still water—of 11·2 feet per second, in order that the water should attain to this speed, it must have a fall of $h = \frac{v^2}{2g} = \frac{11 \cdot 2^2}{64 \cdot 4}$ $= 1 \cdot 95$ feet, so that if the top of the float was just awash when at rest, then, considering the action of one float at a time, and assuming that at the

position of the wheel there was neither a wave crest nor a wave trough due to the motion of the vessel, the water would fall away from the back of the float when the float was in motion for a depth of nearly 2 feet, leaving only $1\frac{1}{2}$ foot immersed, as it would only be at this depth that the speed of the water due to gravity would equal that of the float. How much the denudation of each float will be affected by the action of those in front of and behind it, is very difficult to say.

It is the practice of at least one eminent firm of shipbuilders to make model experiments in a tank, with the wheel in place, and revolved by clockwork at the proper speed, in order to ascertain the amount of the depression of the water-level at the wheel. The usual course followed in designing the wheels for a new vessel is to work from the nearest type within the builder's experience which has given good results. Rankine gives a method of calculating the effective sectional area of a pair of feathering floats, which depends upon the general proposition already stated (page 1), and which is common to all propellers.

If V = speed of vessel in feet per second;
*S = speed of centre of float relatively to the water in feet per second;
A = area of a pair of floats in square feet;
R = resistance of the vessel in lbs.

$$* \ S = \frac{100 \ V}{100 - \text{per cent. of slip}} - V.$$

Then, since $R = \dfrac{W S}{g}$ (see p. 1),

$$R = \frac{64 \times A (V + S) S}{32}$$

$$\therefore \quad A = \frac{R}{2 (V + S) S}.$$

The formula is useful for purposes of comparison, as showing how the effective area varies with power and speed and slip, and it should give a fair approximation to the immersed area of floats at full speed; but as it is not unusual to make the top edge of the lowest float awash when the vessel is at rest, the width in such a case will be greater than that theoretically necessary by an amount equal to the fall of the water-level at the wheel.

In applying this formula to radial wheels, S should be taken as the speed of the lower edge of the float.

In radial wheels the number of floats may equal the number of feet in diameter, and the breadth of a float is usually from $\frac{3}{4}$ of an inch to 1 inch for each foot of diameter. The floats in a feathering wheel should be half as numerous and twice as broad as the floats of a radial wheel. The width of the wheel is generally from one-third to one-half the breadth of the ship. For tugs the breadth of the float may be as much as $1\frac{1}{4}$ inch per foot of diameter.

The diameter of the wheel is determined by the intended speed of the ship, the slip, and the number of revolutions considered desirable, gene-

rally from 20 to 30 per minute, but sometimes as many as 50.

Example :—

Speed of ship 15 knots	**= say 1500 ft. per minute.**
Slip 16 per cent.	**= say 300 „ „**
∴ Circumferential velocity of float	**= 1800 „ „**
Revolutions 30 per minute.	

$$\text{Diameter} = \frac{1800}{3\cdot 14 \times 30} = 19 \text{ feet.}$$

The diameter would be measured at the pivoting points of the floats in a feathering wheel. Fifteen to twenty per cent. is an average slip.

The floats of a feathering wheel are constructed to cleave the water without shock.

Let B D, Fig. 1, represent the direction of motion of the centre of a descending float and the distance moved through by it in a given time if the vessel were stationary, and let D E represent the distance travelled in the same time by the wheel centre relatively to the paddle race, then the actual path of the float relatively to the race is represented by the resultant B E, and the plane of the float entering the water should coincide with this line.

As B D, the angular velocity of the centre of the float, will be less than the velocity of the outer edge, and greater than that of the inner edge, a flat float cannot altogether comply with the conditions required to avoid splashing, but it is approximately complied with if the float is curved to

a radius equal to that of the wheel at the float centres, and the resultant motion of each point on the float as it enters the water will be a tangent

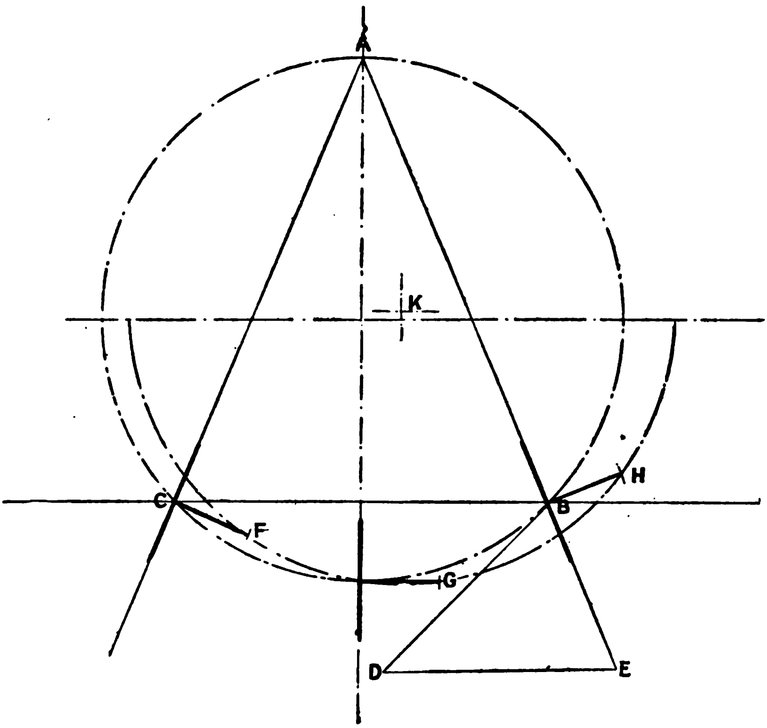


FIG. 1.

to the curve of the float.* The plane of a flat float is a tangent to the centre of a curved float.

A common way of fixing the angles of the floats entering and leaving the water used to be

* See a paper by Mr. A. E. Mills, read before the Inst. of Marine Engineers, 1898–9.

that described by Rankine in 'Machinery and Millwork,' viz. to draw lines A B, A C (Fig. 1) from the summit of the circle passing through the float centres to the points of intersection with the water line B C. Since with this construction B D must equal D E, it assumes that the water arrives at the wheel with the full velocity of feed equal to the radial velocity of the float centres. Although, as before explained, the water is doubtless accelerated to some extent before reaching the wheel, it probably does not attain to the full velocity of discharge until it arrives immediately beneath the centre of the wheel, as it is at this point that the maximum *sternward* velocity of float is reached. It would seem to be more correct to make D E less than B D, that is to say, to make the entering float a little more upright than the construction of Fig. 1 would give. The centre float is usually vertical, although some engineers prefer to incline the top a little aft. The plane of the emerging float, instead of being set off as shown in Fig. 1, is made more nearly vertical in order that the float may act more squarely upon the water throughout the submerged part of its travel. When the planes of the entering, centre and emerging floats are fixed, the centre of the excentric which produces the required motion of the floats can be found. It is the centre K of a circle in which the extremities F, G, H of the float levers lie. These levers vary in length from about one-half to three-fifths of the

depth of the float to which they are fixed perpendicularly, or nearly so, and usually a little below the centre. The shorter the levers, the more upright will be the position of the float when entering and leaving the water for a given excentricity of the driving centre. This excentricity varies in recent practice from 7 to 8 per cent. of the diameter of float centres.

There are so many uncertain quantities in the problem that it is not possible to construct very precisely the proper float angles. Departure from the designed speed of ship, rate of revolution or immersion of wheel will throw out the angles of entering and emerging floats, but probably the efficiency will not be greatly affected should these depart somewhat widely from the theoretical positions. Most engineers have their own ideas as to the laying off of a wheel, and there is considerable divergence in practice.

The following construction for a wheel with curved floats should give good results, but it is doubtful if in any case a much greater excentricity than 8 per cent. is desirable.

Let A B C (Fig. 2) be a circle drawn through the lower edge of the entering float. Let B D represent the tangential velocity of the float edge, and let D K represent the velocity of the vessel. Produce D K to E, making D E equal to B D. K E will then represent the slip of the wheel periphery. Bisect K E in L and draw L B. With a

radius equal to that of the float centres draw the curve of the float so that L B is a tangent to it at B.

At C′ draw the tangent C′ G = D B and G H parallel to B C and equal to D E. Join C′ H. With a radius equal to that of the float centres

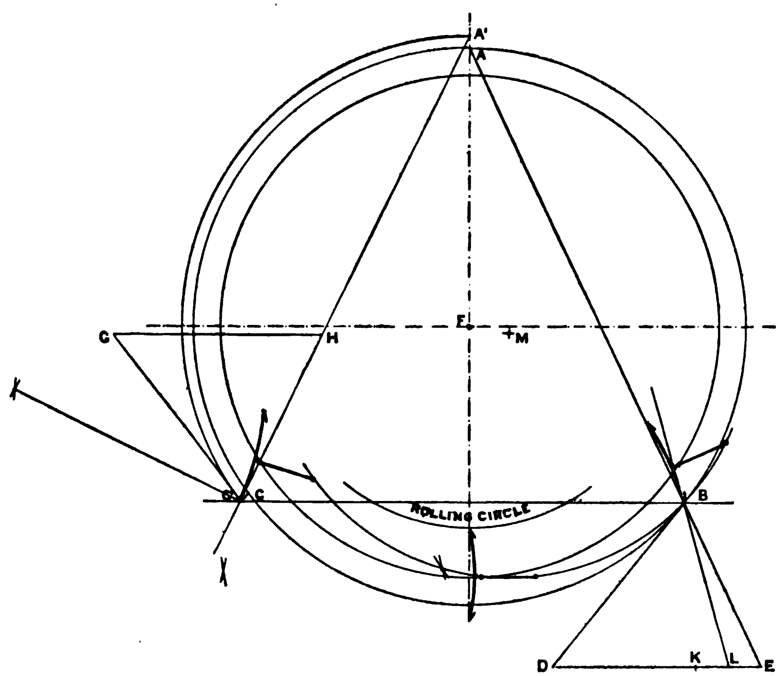


Fig. 2.

draw the curve of the float so that C′ H is a tangent to it at C′. Let the chord to the centre float be vertical. Make the length of the levers three-fifths of the depth of the float and find the centre M of the circle passing through their extremities.

With this construction the entering float will have the inclination appropriate to a velocity of feed equal to the speed of the ship plus one-half the maximum acceleration imparted by the periphery, and the emerging float leaves the water at an angle appropriate to the full velocity of discharge. Throughout the underwater part of the travel the action is as nearly sternwards as is practicable.

Mr. Mills points out in the excellent article already referred to that no part of the float should enter the rolling circle, which is a circle at which the radial velocity is equal to the speed of the vessel.

A feathering wheel does not produce a uniform acceleration of the whole of the water acted upon. The acceleration reaches a maximum under the centre of the wheel where the sternward velocity of the float is greatest, and as the float pivot ascends with a constant angular velocity, but with a diminishing sternward velocity, the water which is still in contact with the float is retarded again.

Even the best feathering wheel cannot altogether comply with the condition laid down by Rankine as that which should be found in a perfect propeller, namely, that the water should enter the propeller with a velocity of feed equal to V and be gradually accelerated up to the velocity of discharge V + S, the mean speed of the propelling

instrument being $\frac{V+S}{2}$ and its efficiency $= \frac{V}{V+\frac{S}{2}}$. To accomplish this it would be necessary that the angular velocity should increase as the floats travelled aft. Mr. R. E. Froude has suggested that as the water is piled up by the driving face of the float it will leave the wheel with a certain amount of "head," and will finally attain a velocity in the race greater than the speed of the wheel, and that therefore the efficiency may approach more nearly to $\frac{V}{V+\frac{S}{2}}$ than is at first sight apparent.

There is less splashing and less waste of work in imparting downward and upward motions to the water as compared with a radial wheel, but it cannot be considered a perfect propeller.

Fig. 3 represents the wheels of a vessel 260 feet in length, 28 feet broad, and 5 feet 9 inches draught of water. The I.H.P. was 2680 and the speed 18¼ knots. The diameter of the wheel was 20 feet 6 inches over the floats, which were made of elm 9 feet 9 inches by 3 feet 6 inches. The revolutions were 47 per minute and the slip 26½ per cent. The lowest float was immersed one inch from the top edge when at rest. The slip is rather high and it is probable that a better per-

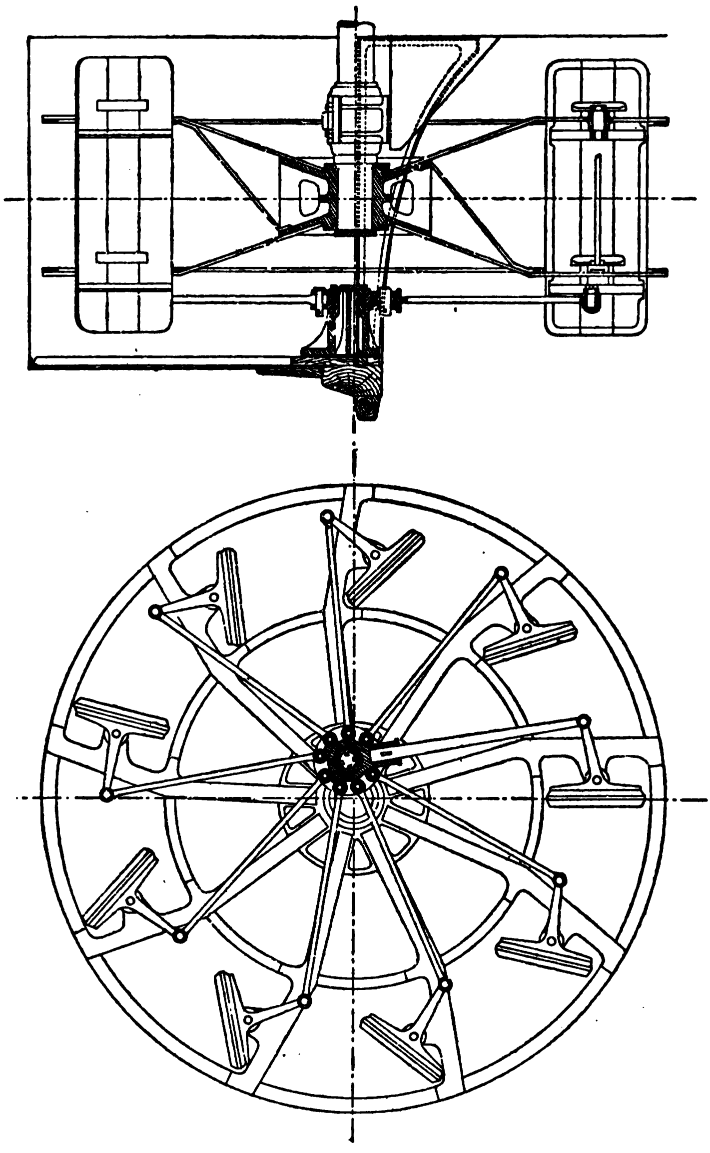


Fig. 3.

formance would have been obtained if the wheel had been rather more immersed. Fig. 4 shows a wheel with curved floats fitted in a vessel 230 feet in length and 27 feet 6 inches broad. The I.H.P. is 2520 and the speed 18¼ knots. The diameter

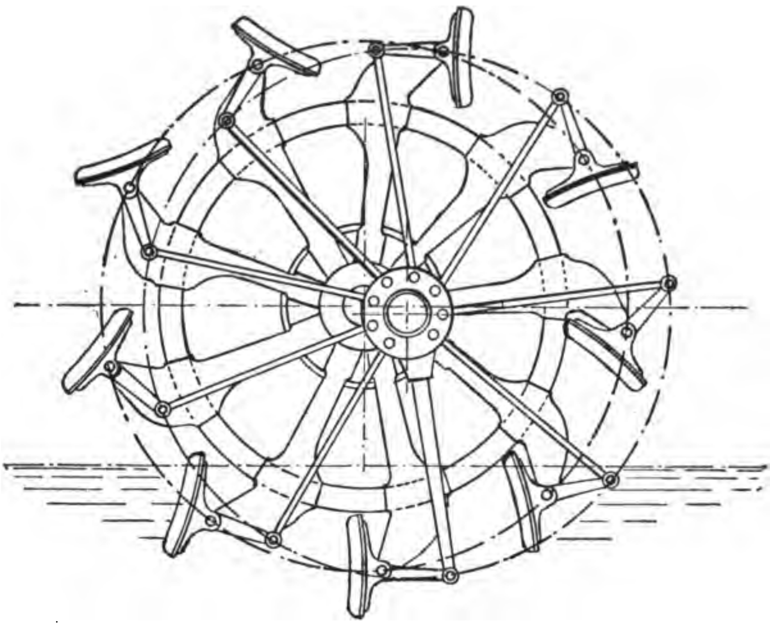


FIG. 4.

of the wheel is 20 feet over the floats, which are 9 feet 6 inches by 3 feet 7½ inches. The revolutions are 41 per minute, and the slip 15½ per cent. The top of the lowest float is immersed 18 inches at load draught.

These two wheels being fitted on somewhat

similar vessels of the same speed, and being driven by engines of about the same power, compare very well together. The curved floats more deeply immersed appear to give considerably less slip than the wheel shown in Fig. 3.

CHAPTER III.

THE SCREW.

THREE strings wound spirally round a cylinder make a three-threaded screw. If instead of strings flat blades be wound edgewise, each having an edge soldered to the cylinder, then if a slice be cut off as shown in Fig. 5, there will be one piece of blade attached to the slice if the screw have one thread, two pieces if two threads, and so on.

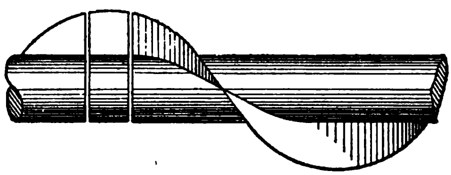


FIG. 5.

The "length of the blade" is equal to the length of the slice thus cut off, and the length of the cylinder necessary to contain one complete convolution of the blade is the "pitch" of the screw. The ratio of the length of the blade to the pitch is called the "fraction of pitch," a term more in use on the Continent than in this country.

The face of the blade which presses against the water when propelling the vessel ahead is

called the "front" or "driving face"; the opposite face is the "back" of the blade. When speaking of the screw as a whole the nomenclature is commonly reversed, "in front of the propeller" standing for forward of the screw in relation to its position on the vessel, and "behind the propeller" meaning abaft it. These terms "in front" and "behind," which are apt to be confusing, will be avoided in this work.

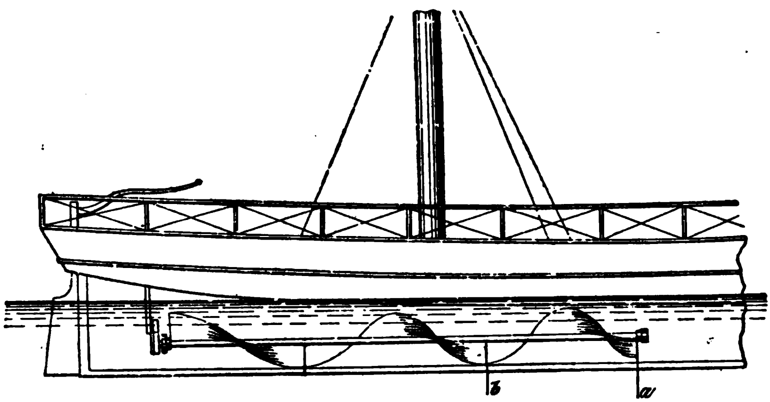


FIG. 6.

Fig. 6 shows a screw of "expanding" or "increasing" pitch as originally proposed by Woodcroft. The length of cylinder necessary to contain a complete convolution is not the same at all parts, and the pitch therefore is not "uniform." The first turn of the thread has a pitch equal to the length *a b*; the second has a longer pitch equal to *b c*. If a slice be cut off the cylinder (repre-

sented in Fig. 6 by the screw shaft) midway between a and b, and of any length, it would be said to have a mean pitch equal to $a\,b$. A slice taken at a position midway between b and c would be said to have a mean pitch equal to $b\,c$, and similarly for any other part.

The edge of the blade which cuts the water when driving the vessel ahead is called the "forward" or "leading" edge, distinguishing it from the "after" or "trailing" edge.

The "disc area" is the area of the circle swept by the blade tips. Rankine speaks of "effective disc area," which is the area as just defined *minus* the area of the boss, but it is not now usual to make this deduction when dealing with disc areas.

The "developed area" or "blade surface" is the sum of the areas of all the blades, exclusive of the boss.

The "projected blade area" is the sum of the areas of the blades, exclusive of the boss, projected upon an athwartship plane.

It is convenient to use a term expressing the relation which the product of the pitch and revolutions of the screw per minute bears to the advance of the vessel through the water per minute. This term is "apparent slip."

If P = mean pitch in feet;
R = revolutions per minute;
V = speed of ship in feet per minute;

then $\frac{P\,R - V}{P\,R} \times 100$ = percentage of apparent slip.

In the great majority of cases P R is greater than V, but it is occasionally equal to or less than V. The apparent slip is positive, nil, and negative in the three cases respectively.

The real slip or acceleration of the water S (see p. 1) is not measurable from the pitch and revolutions of the screw. A propeller placed at the stern of a vessel is in a current having usually a forward mean velocity U relative to still water, caused by the friction of the skin of the vessel.

While the vessel advances at a velocity V through the water, the screw advances at a less velocity = V − U. If the apparent slip is nil then the real slip of the screw is equal to U.

Conceive a propeller of increasing pitch which shall be so designed that the pitch of the forward edge multiplied by the revolutions per minute shall be equal to the speed of the propeller through the water, that is V − U; and the pitch of the after edge multiplied by the revolutions equal to V, then the mean pitch, as already explained, would be called $\frac{V + (V - U)}{2}$. The late Mr. Froude showed * that under these circumstances apparent negative slip is possible. He describes an ideal case in which the whole of the resistance

* Minutes of Proc. Inst. Civil Engineers, xxxii. 1870–71.

of a vessel consists in skin friction, wave-making and other factors being excluded. The dynamic equivalent of the propulsive force employed in keeping her in motion is found in the frictional wake, and a propeller which should pervadingly operate upon the wake in such a manner as to bring it gradually to rest would, in thus neutralising it, maintain the propulsive force, and, given established motion, a theoretically perfect propeller, quite clear of the ship's stern, would maintain that motion and exhibit apparent negative slip equal to half the forward mean velocity of the wake at the point where the propeller operated.

In this case it is clear that apparent negative slip results from the fact that while a sternward velocity is supposed to be imparted to the water equal to the product of the number of revolutions multiplied by the pitch of the after edge of the screw, the mean pitch of the screw itself is nominally less than the pitch of the after edge. Negative slip would disappear in this particular case if the speed of advance of the screw were calculated from the after pitch instead of from the mean.

Apparent negative slip is sometimes exhibited by screws of uniform pitch when the ratio of pitch to diameter is small.

The twin screws of H.M.S. *Collingwood*, with a pitch ratio of 1·5 gave 1·26 per cent. apparent negative slip, and this was increased to 2·56 per cent. when the pitch ratio was reduced to 1.

Numerous explanations have been given of the phenomenon of apparent negative slip, but none of them can be accepted as satisfactorily accounting for its occurrence in the case of well-formed uniform-pitch screws, the pitch of which has been carefully verified as in the case of the *Collingwood.*

It has been suggested that the blades twist or spring under the pressure of the water, which would have the effect of increasing the pitch, and that they recover their shape when the pressure is relieved, so that measurements taken after the trial even would be misleading.

If this were the case, screws with thin blades would be most likely to show negative slip, but it is, on the contrary, generally met with in screws with very thick blades where springing would be least likely to occur. Moreover, it might be expected that the metal would soon give way by fatigue if it were distorted by the pressure, as the fluctuations are very great and very rapid.

What has been said above about the effect of thick blades lends force to another suggestion, which is that negative slip may be attributed to the effect of the round back of the blades. The pitch is measured on the assumption that both faces of a blade are alike. The effect of the round back of the blade must be to increase the effective pitch and to tend to reduce the apparent slip,

although it is difficult to say by how much, and it does not appear to be sufficient to account for the *Collingwood* results.

It has been shown that the intermittent action of the blades passing through the dead water abaft the stern-post of a full-formed ship reduces apparent slip and may even cause it to change sign, but this does not apply in the case of the ship referred to, which is a twin-screw ship, and is of a fine form.

Another suggestion has been that the motion of the particles in the wave which followed and enveloped the stern of the ship might produce apparent negative slip. All attempted explanations based upon the effect of dead water and following current alone, apply only to a screw of increasing pitch, and this has been already dealt with (p. 24). They are quite inapplicable to a screw of uniform pitch, and the same may be said of the theory based upon the circular motion of the particles of water in waves. The waves referred to are presumably those made by the vessel, because it cannot be contended that negative slip results from the screw working among free waves. It is well known that the contrary effect is produced, the slip is increased, and the most favourable condition for the exhibition of apparent negative slip is still water. If a wave follows the ship, and its energy can be made use of in any way by the screw, that wave has pre-

viously been created by the ship from which the energy has been robbed, and can only be partially restored. Could it all be given back, and could the whole energy of the frictional wake be utilised by the screw without loss, there would still be no surplus thrust to balance the waste of energy caused by diverging waves and to keep the vessel in motion. It is certain that there must be a stream of water left behind by the screw, having sternward motion relative to still water. How is it then that notwithstanding this necessity, apparent negative slip is occasionally obtained with screws of uniform pitch? In a paper read before the Institution of Civil Engineers on the Screw Propeller,* the author gave what seems to him a satisfactory explanation based upon a proposition recently enunciated by Mr. R. E. Froude,† to the effect that the slip or acceleration S of the water in the race was always in excess of the slip of the screw P R − V.

Mr. Froude showed that if no rotation were produced in the race by the propeller, a limiting case was reached in which one-half of the whole acceleration would be produced forward of the propeller and one-half aft of it. The water forward of the propeller was affected before it was actually in contact with it, and would run towards the screw, meeting it with a velocity as regards

* Proc. Inst. Civil Engincers, cii. p. 74.

† Trans. Inst. Naval Architects, 1889, xxx. p. 390.

still water, of $\frac{S}{2}$, and the action of the propeller upon the water while in contact with it was to accumulate pressure by elevating the surface over the screw, which had the effect of increasing the acceleration of the race after it had left the propeller.

This is perhaps more easily understood if we suppose the propeller P, Fig. 7, to be stationary in an ocean moving with a velocity V in the

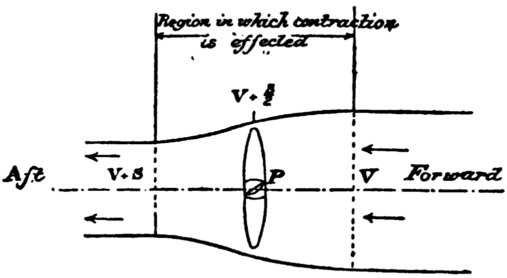


FIG. 7.

direction of the arrows. At some distance forward of the screw the water will be advancing towards it at a velocity V. On nearing the propeller, the water is accelerated by its sucking action, and meets it with a velocity $V + \frac{S}{2}$.

The length of the blades may be supposed to be so small that no appreciable change of velocity can take place in the stream while actually passing through them, but after leaving them the speed of the stream is further accelerated up to the final speed $V + S$. The mean speed of stream in which such a propeller works is therefore $V + \frac{S}{2}$, and the

real slip of the propelling apparatus, which is a measure of its efficiency, would be $\frac{S}{2}$, but the speed of the race is $V + S$ and the total acceleration or real slip of the water is S. Such a propeller might show a considerable amount of apparent negative slip if placed behind a ship and in a following current, a condition which is of course essential, because if it were propelling a phantom ship, that is a ship which leaves no wake (see p. 81) the apparent slip would be the same as the real slip, viz. $\frac{S}{2}$. It is impossible to make an open screw which shall not rotate the water, but the finer the pitch is the less the rotation will be.

Mr. Thornycroft has shown * that the relation between the amount by which the race is accelerated forward and aft of the screw respectively may be expected to depend upon the amount of the rotation produced. A screw of coarse pitch-ratio which will rotate the race considerably, will produce a large proportion of the whole acceleration before the water reaches the screw, leaving only a small part to be imparted abaft it. When the rotation is a maximum the whole acceleration is produced by suction, and the speed of the stream on meeting the propeller is $V + S$, in which case the real slip of the screw is equal to the acceleration of the water S. All open propellers, by which

* Trans. Inst. Naval Architects, 1889, xxx. p. 419.

is meant those which are not confined in a casing like the screw-turbine, occupy some intermediate position in our imaginary column, as indicated in Fig. 8, and are working in a stream with a velocity varying between $V + S$ and $V + \frac{S}{2}$, depending upon the greater or less rotation of the race. Hence the finer the pitch ratio, the more favourable would be the conditions for obtaining

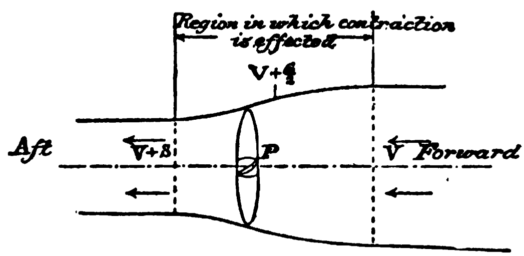


FIG. 8.

apparent negative slip, and it is found to be invariably the case that it only occurs under these conditions, and that it may be increased by still further reducing the pitch of screws exhibiting it (see p. 24).

If the pitch ratio is so fine as to allow the propeller to approach the position shown in Fig. 7, the loss by surface friction becomes excessive, and instead of the efficiency being increased by the reduced slip it is diminished. In practice there is a point beyond which any gain by reduced rotation of race is counterbalanced by a loss

through increased friction. This point probably lies between pitch ratios 1·0 and 1·5, and is reached before the pitch ratio becomes sufficiently fine to produce apparent negative slip, and such a result should never be sought. As a matter of fact, it is probable that a propeller could be designed for any ship which should give apparent negative slip, and from the Table, p. 108, not only the proportions for such a screw, but also the approximate amount of the negative slip can be calculated and the efficiency of the propeller estimated. The constants in the Table giving apparent negative slip are printed in small characters, and the low efficiency to be expected will be seen at a glance. This Table will be fully explained in the next chapter.

It follows from Mr. Thornycroft's reasoning given above, that not only is the efficiency of the screw less if the pitch ratio is very great, but a loss of power will also be caused by an increase in the resistance of the vessel which it is propelling if the screw is placed in such a position that the suction from it, which will extend some distance ahead of it, can take effect upon the hull. This is probably the reason why fine-pitched screws give the best results in full-lined cargo vessels. It is a common practice to make the pitch of screws for such ships about equal to the diameter, and probably the suction from a coarse-pitch screw would considerably augment the hull resistance.

MEASUREMENT OF PITCH.

There are a number of pitchometers made which will measure the pitch of a screw with sufficient accuracy if it is uniform. They are not to be depended upon for the measurement of increasing pitches.

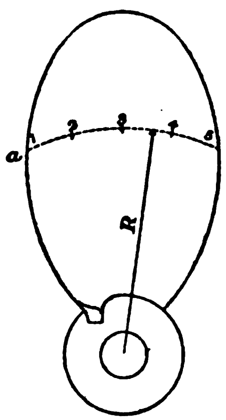


FIG. 9.

The operation may be performed without instruments as follows:—Strike an arc of a circle ab, Fig. 9, concentric with the axis of the propeller upon one of the blades with any radius R. Divide the arc into a number of equal intervals as 1, 2, 3, 4, 5. Measure off the same number of intervals upon a base line x y, Fig. 10, making them equal to the developed length of the intervals

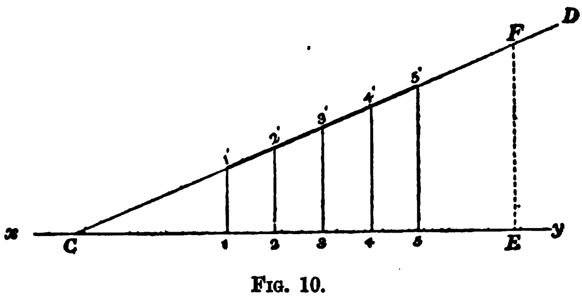


FIG. 10.

upon the arc. Measure the ordinates at each interval from a plane perpendicular to the axis of

the propeller, and lay off the ordinates 1 1′, 2 2′ 3 3′, &c., at the corresponding intervals on the base line. Draw a line C D through the points 1′, 2′, 3′, &c., and produce it to cut the base at C.

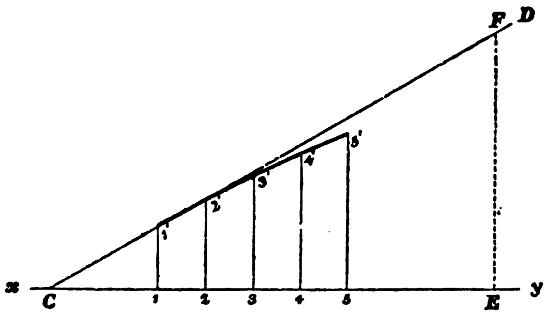


FIG. 11.

If the pitch is uniform C D will be a straight line. Measure off from C a length C E, equal to the circumference of the circle of R radius, and erect a

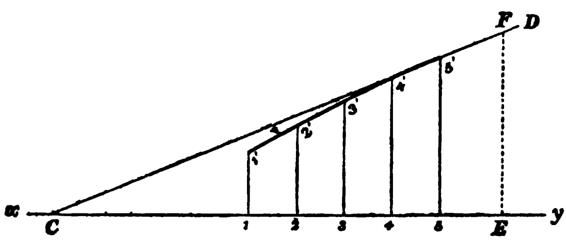


FIG. 12.

perpendicular at E cutting C D at F. E F is the pitch of the screw. If the pitch is not uniform the points 1′, 2′, 3′, &c., will lie in a curve. Tangents must be drawn to the curve at the extremi-

ties 1′ and 5′, and the pitch of each measured separately, see Figs. 11 and 12, and the mean of the two will be the mean pitch of the screw. Measurements should be made of each blade, and at a number of different radii, and the mean of all the readings taken as the mean pitch.

If the pitch is uniform, and great accuracy is not required, choose any two points on the arc *a b*, Fig. 9, such that radial lines from them to the axis subtend an angle of 30°. The difference between the length of the ordinates of these points from a plane at right angles to the axis, measured in inches, is equal to the pitch in feet.

The pitch of small model screws used for experimental purposes can be most readily obtained as follows:—Make a cylinder of wood of a diameter about two-thirds that of the propeller and 2 or 3 inches in length. Fit a short mandril to represent a portion of the shaft into the propeller boss, and pass the mandril through a hole in the axis of the cylinder which has been bored to fit it. Place the cylinder upon a table with the axis vertical, and wrap a sheet of paper round the cylinder, securing it with an elastic band, and cut the edge accurately to fit the face of the propeller blade. The direction of the axis F E, Fig. 13, should be marked upon it. When the paper is removed from the cylinder and unrolled, the edge which fitted the face of the blade will form a straight line as C F if the pitch is uniform, and if D be

the diameter of the cylinder, then $\frac{FE}{CE} = \frac{\text{Pitch}}{D \times \pi}$. If the blade is not of uniform pitch it will form a curved line, and the pitch of the leading and after edges must be obtained by drawing tangents to the

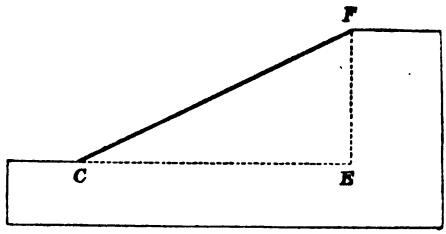


FIG. 13.

extremities of the curve as already described.* It is sometimes useful to be able to estimate roughly the pitch of a screw at sight. This may be done by observing at what radius the blade makes an angle of 45° with the axis; the pitch is equal to the circumference at this radius.

VARIOUS FORMS OF SCREWS.

The screw was brought into successful operation as a propeller by Ericsson and Smith in 1836. The *Archimedes*, a screw vessel of 237 tons burden, was built by the latter in 1839. The screw used

* See also a very good method of measuring the pitch of model screws, described by Prof. Durand, Trans. American Society of Naval Architects and Marine Engineers, vol. v. p. 115.

was a single-threaded helix of one complete convolution. A double thread of half a convolution was afterwards tried, and found to be an improvement, but the best result was obtained with two threads and one-sixth of a convolution.

The Earl of Dundonald in 1843 patented a propeller with the blades thrown back as shown in Fig. 14, the object being to counteract centrifugal motion of the water supposed to be caused by the rotation imparted to it by the screw.

When a propeller is not sufficiently immersed

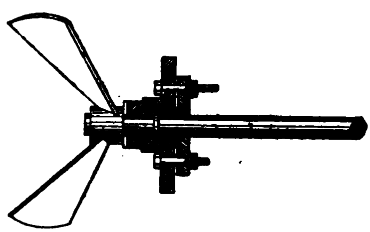


FIG. 14.

to prevent it from drawing down air, it is probable that centrifugal action takes place, and that the column of water takes the form of a cone with the screw for an apex, but when no air gets to the propeller no such dispersive action takes place, the column on the contrary contracting as it passes through the screw as shown in Fig. 7.

In 1849 Robert Griffiths patented a self-governing propeller, which he thus describes: "If the screw moves with greater velocity than usual, the increased resistance of the leading edge

shall correspondingly increase the pitch, thus increasing the resistance and bringing down the revolutions."

The propeller chiefly associated with his name is shown in Fig. 15, which represents the form still largely used in the British Navy. The principal feature in the Griffiths screw is the large boss, which, while not impairing the efficiency, enables the blade to be fixed in such a manner that the pitch can be readily altered. This is an important consideration, as it is difficult to fix upon exactly the right pitch in designing a propeller to run at a given number of revolutions.

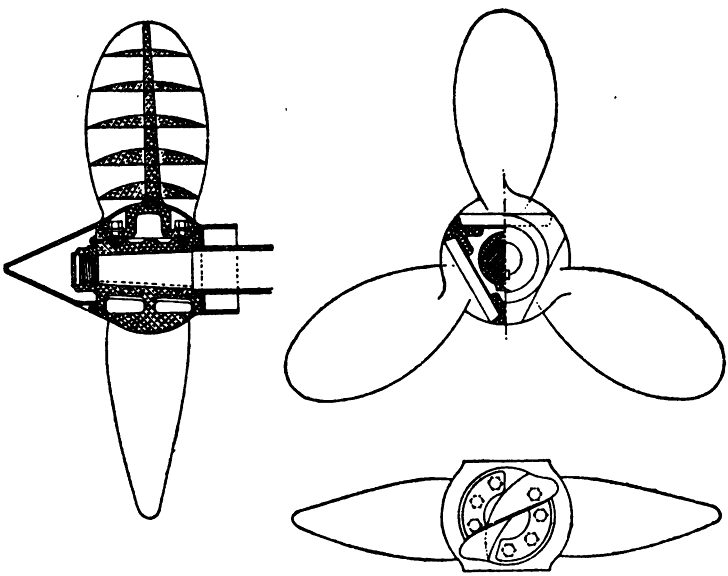


FIG. 15.

The Hirsch screw is shown in Fig. 16. It has an increasing pitch, and the propelling surfaces are so formed as to throw the water somewhat towards the axis.

The Mangin propeller, Fig. 17, consists of two narrow-bladed screws set one behind the other on the screw shaft with a space between them.

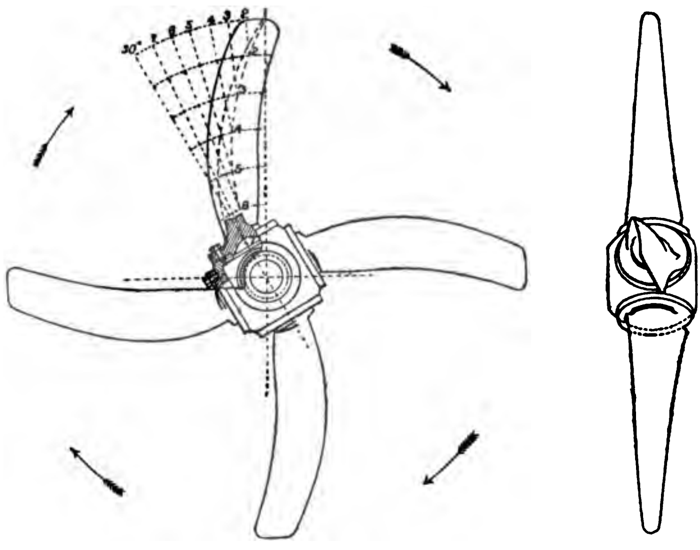


FIG. 16.

It is supposed not to rotate the water so much as other screws.

Rigg's propeller had fixed guide-blades placed astern of the revolving screw, the guide-blades being set at such an angle as to take the rotation out of the water and leave it moving directly astern.

Screws have been tried with the pitch near the boss less than the pitch at the circumference, so as to allow the central part simply to follow up the water without propelling.

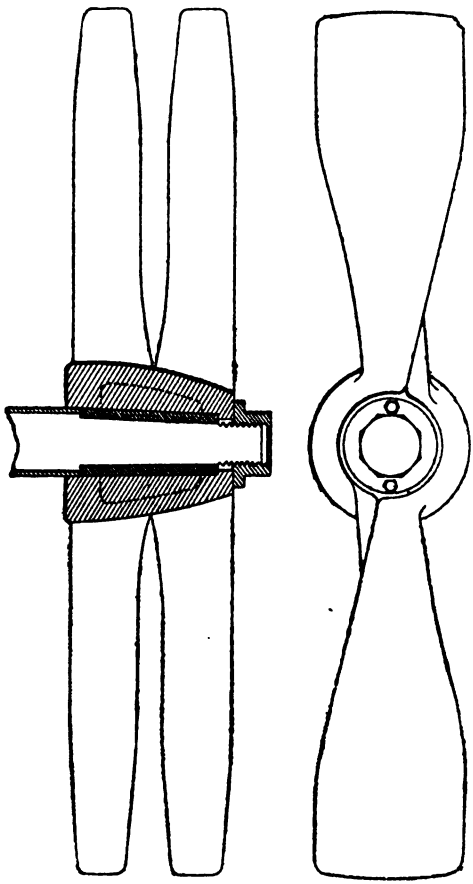


FIG. 17.

The Thornycroft screw, Fig. 18, which has proved very successful, has an increasing pitch

from the forward to the after edge at the middle of the blade, but the pitch becomes uniform at the root and at the tip, the reason being that at the root it is not advisable to increase the rotation of the water by increasing the pitch, and if it is attempted to accelerate the water too much at the tip, it escapes round to the back of the blade.

The blades are thrown back after the fashion of the Dundonald propeller, but instead of being

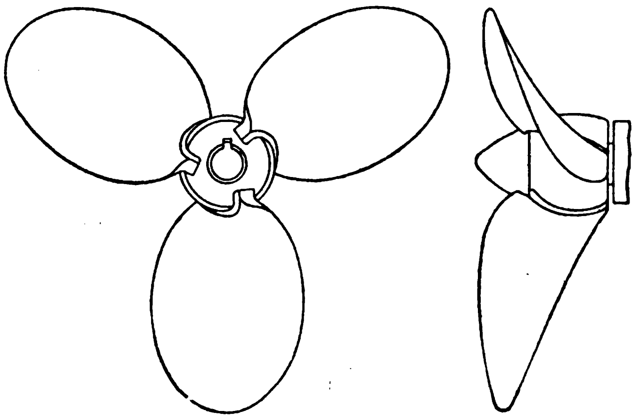


FIG. 18.

straight they are convex on the driving face. This has the advantage of getting the blades further from the vessel's stern for a given length of screw shaft, and they act more squarely upon the contracting column of water passing through them. The propeller shown in Fig. 18 has forged steel blades keyed into a forged or cast-steel boss. The form of the key used is shown in Fig. 19. Manganese, or Stone's bronze has very generally

taken the place of forged steel, the screws being cast in one piece.

The scantlings can be made as thin as steel, and the surface is smoother and remains in better condition on service. The proper helical shape

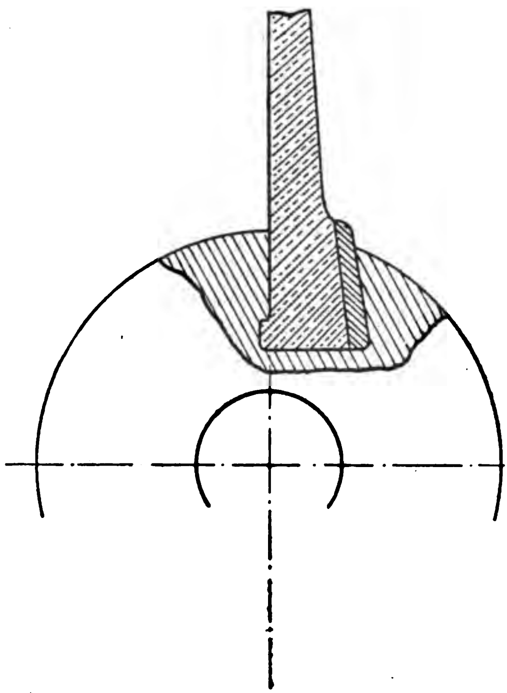


FIG. 19.

of the blade can be carried down to the root in a screw which is cast solid; but wedged blades are necessarily flattened where they enter the boss, and the wedges themselves destroy the smoothness and regularity of form so desirable at high rates of revolution.

The Thornycroft screw-turbine propeller will be described in a separate chapter, see p. 173.

An ingenious attempt to make use of a certain amount of energy said to be wasted by the screw has been made by MM. des Goffes and de George.

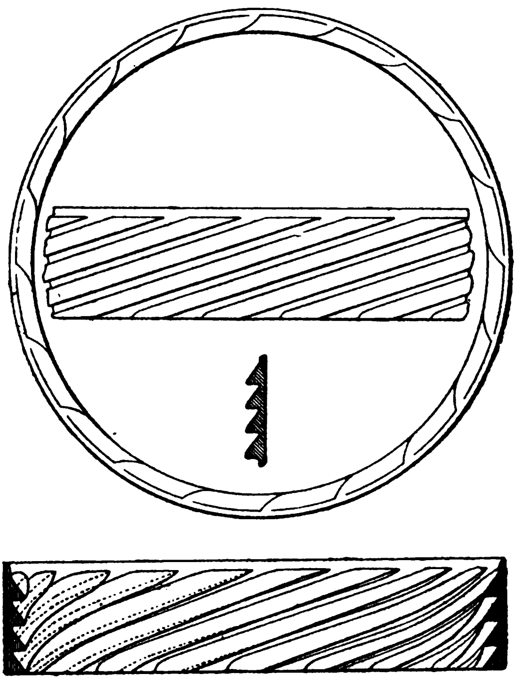


FIG. 20.

The inventors say that the viscosity or resistance to rupture of the water in which any helical propeller revolves, causes an appreciable current to be set in rotation just beyond the tips of the screw blades. It is contended that a series of

helical surfaces, opposed in direction to those of the screw proper, will receive a thrust from the revolving ring of water which can be utilised for propulsion. The "Antispire," as it is called, can be placed around a screw propeller of any form. It is shown in Fig. 20. The only experiments with the apparatus which the author has witnessed were made in a tank, and as there was no motion through the water it was not possible to tell how much the result would be affected by the friction of the ring itself; but a large increase of thrust was shown to be produced by it, and it is possible that the friction of the surfaces would be more than counterbalanced, and a real propulsive force exerted. It is reasonable to suppose that there is an unutilised reservoir of work in revolving currents external to the propeller disc. Rings or bands without helical blades have been used for protecting screws and for giving increased manœuvring power, but these plain rings do not increase the thrust or the efficiency of the screw, and the addition of the blades would certainly be found advantageous in such cases, and may be worthy of a still wider application, as, for example, to tugs in which the speed through the water is small.

Position of the Screw.

In some double-ended ferry boats, both in this country and America, screws have been placed

at both ends of the vessel for what appear to be sufficient reasons connected with the service which they have to perform. In the well-known Mersey ferry boats there are four screws, but in some of those built in America two only are employed, one forward and one aft, driven by the same shaft, an arrangement which appears to be inferior. The forward screw of one of these latter vessels was estimated to augment the resistance of the hull by 23·5 per cent., and its propelling efficiency was only 43 per cent. of that of the after screw.

It was said on p. 23 that a propeller placed at the stern of a vessel was in a current or wake having *usually* a forward velocity. By the wake of a vessel is meant the water affected by the passage of the vessel herself, apart from the effect produced by the screws. This wake may be divided into three components which together determine its character. The stream-line action tends to produce a following current which may be called the stream-line wake; the friction of the vessel's skin also creates a following current or frictional wake; and there is besides an accompanying system of waves.

The direction of the wave-motion wake in relation to the propeller depends upon whether the latter falls under a crest or a hollow. If the propeller be under a wave crest the motion of the particles of water will be forward, if under a

hollow it will be sternward. The resultant wake will be dependent upon the relative magnitude of each component.

The frictional and stream-line wakes always tend to reduce apparent slip, and even when the wave-motion wake is opposed to them it is not usually sufficiently strong to neutralise them. But in some exceptional cases they are more than neutralised, with the result that the wake is actually negative in value, that is, it has a sternward motion in relation to still water, and the real slip, when such is the case, is less than the apparent slip. Some of the torpedo-boat destroyers have a small negative wake at the propellers, but such cases are very rare. As a rule the wake has a forward velocity due chiefly to the friction of the skin of the vessel, which may be taken as averaging 10 per cent. of the speed. It is affected more by the nature and extent of the surface than by the form of the ship.

If the screw works in this wake it is able to recover some of the energy which has been expended by the ship in giving it motion. It would not be desirable to increase the volume or the velocity of a wake for the purpose of improving the efficiency of the propeller, because surface friction is responsible for the greater part of the resistance of a ship at moderate speed; but as it is a necessity that there should be a wake, it is a distinct advantage to place the propeller in it

and allow it to utilise as much as possible of the energy it finds there.

This frictional wake must not be confused with dead water, which is water eddying behind a bluff stern, and which has acquired the full velocity of the vessel. When once the speed of the vessel has been imparted to this water not much energy is wasted in maintaining it. If it is drawn out by the screw, fresh water must take its place, and there will be a continual drain of energy from the ship, as the inflowing water must in its turn have the full forward velocity imparted to it. Dead water is almost a thing of the past, and is met with only in the case of very full ships.

If a screw is placed behind a stern so bluff that the supply of water is impeded, it will draw in water at the centre of the driving face and throw it off from the tips of the blades like a centrifugal pump. It is recorded that an attempt to propel a square-ended caisson by means of a screw resulted in the caisson going astern, whichever way the screw was driven.

It is very important that a propeller should have sufficient immersion, since if it breaks the surface of the water, its efficiency may be reduced to a remarkable extent (see Plate I.) by reason of air which is drawn down, and the greater the depth of the screw below the surface the less is the chance of its being thrown out by pitching or rolling. The speed with which water can follow

up the blades of a screw depends upon the head over them, but when air is excluded the equivalent of a head of 30 feet of water is supplied by the atmosphere, and this being elastic and having practically no inertia to resist sudden motion, its pressure is more effective than that of a column of water of equivalent weight.

There is a disadvantage connected with an inclined screw shaft which points to the advisability of placing the shaft as nearly horizontal as possible.

The result of depressing the end of the shaft is to cause the effective pitch to vary through every part of the revolution. If the inclination be supposed to be 45° for example, that part of the blade which is intended to have a pitch of three diameters has in reality an effective pitch varying from nothing to infinity.

It is of course obvious that the pitch of the blades in relation to the axis is unchanged by any alteration in the direction of the shaft, but whatever the pitch in relation to the axis may be, if the axis were to pass vertically out through the bottom of the ship, the virtual or effective pitch, measured in the direction of motion is nil. If a screw does not move along but has a motion of rotation only, the resistance of the water to the blades is the same whatever be the direction of the shaft; but if the propeller be allowed to move forward, while at the same time it be constrained

to move horizontally, the shaft being inclined to the horizontal, then the resistance of the water to the blades is not uniform, but varies over every part of the revolution. This will perhaps be made clearer by an examination of the phases through which a blade passes during one revolution. It is convenient and suitable to consider the action of a screw as similar to that of an inclined plane moving past the stern. In Fig. 21 the full line represents the upper blade as a plane moving from port to starboard; the dotted line represents the lower blade as a plane moving from starboard to port. In Fig. 22 the shaft is horizontal, and the full line shows the blade going down, and the dotted line the blade coming up. In Fig. 23 the shaft is inclined at 45°, the full line again showing the blade going down, and the dotted line the blade coming up. As the ship moves forward the water may be supposed to flow to the screw in approximately horizontal lines, and the blade which at one part of the revolution is edge-

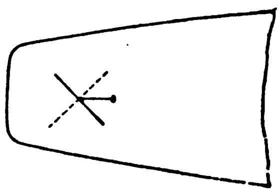


FIG. 21.

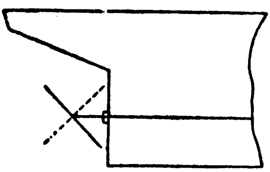


FIG. 22.

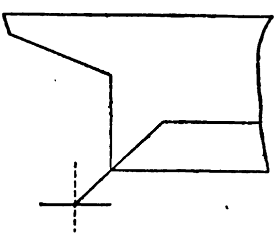


FIG. 23.

wise to the water, at another is square on to it, and the result is an irregular pressure causing vibration. Another way of looking at it is this: A particle of water meeting the ascending blade has its motion relative to the vessel arrested completely, while a particle on meeting the forward edge of the descending blade would require to have its velocity infinitely accelerated in a horizontal direction, to enable it to escape from under the blade. This is what is meant by saying that, in the above example, the effective pitch varies from nothing to infinity during each revolution.

Wide-Bladed Screws.

When the length of the propeller in the direction of the axis is small, that is, when the blades are narrow, there is probably not much to be gained by departing from a true helical surface, or what is called a uniform-pitch; but when the blades are wide the pitch should increase in the direction of the length of the propeller, that is, the after edge of the blade should have a coarser pitch than the forward edge. The reason for this may be seen by referring to Fig. 8. The column of water passing through the screw is contracting in area and increasing in velocity.

Blades of uniform-pitch would only be strictly appropriate if the column while passing through

the screw were parallel and maintained a constant speed.

If the length of column occupied by the screw is sufficient to allow a sensible contraction to take place within its limits, then the pitch of the screw surface should augment at the same rate as the speed of the column of water is accelerated, in order that all parts of the blade may keep touch with it during its passage. It was an early practice introduced by Woodcroft to vary the pitch in this manner (see page 21), the supposition being that by so doing a gradual acceleration would be produced and not a sudden one. It is probable that in no case could water be accelerated suddenly by a submerged propeller, and all that is required is that the surface of the screw should be adapted as nearly as possible to the rate of flow through it, which rate is determined by the mean pitch of the screw surface. What the variation on each side of the mean should be is very difficult to say, as it has not yet been determined at what distance ahead of the screw acceleration of the water commences, or at what distance astern it is completed, and the full velocity of race attained.

Although we know that the *vena contracta* of the race must be somewhat of the form shown in Fig. 8, it is not possible at present to define its boundaries, and it can therefore only be stated in general terms that the greater the slip ratio the

greater would be the contraction, and consequently the greater should be the variation of pitch on each side of the mean. Since slip ratio at a given efficiency increases with pitch ratio, the variation should also bear some proportion to the pitch ratio. As the use of wide blades is frequently associated with high slip ratio, as for example, when diameter is restricted by the draught of water, not only do they occupy a considerable length of the contracting column, Fig. 8, but also the amount of the contraction is greater; and if this reasoning is correct, there is a twofold advantage to be gained by giving an increasing pitch to screws with wide blades.

Vibration.

In order to prevent vibration from being set up by the propeller in long fine vessels of high power, two things should be considered. The propeller should have a good running balance, and the centre of pressure should be in the centre of the disc. To ensure that the first condition is realised, each blade must be of the same weight, and the centre of gravity of each must be at the same distance from the axis of the shaft. To satisfy the second condition is more difficult. If the screw works in undisturbed water and the surface of each blade is disposed symmetrically about the shaft, then the centre of pressure will

be in the centre of the disc if the screw is caused to advance in the direction of the line of its shaft.

Any inclination of the shaft from the line of advance tends to throw the centre of pressure out of the centre of the disc, for the reason already explained (see page 48), and the same effect is produced to some extent by the inequality of the onward motion of the water in the frictional wake in which the propeller works.

It is possible to get these two disturbing elements to counteract one another to some extent. Thus a slight inclination of the shaft from the line of advance in a horizontal direction will correct the effect due to different velocities of stream at the top and bottom of the propeller disc; and in twin-screw vessels, where the forward velocity of the frictional wake would be greatest nearer the side of the ship, the tendency to throw the centre of pressure inwards can be counteracted by a slight inclination of the shaft in a vertical plane. Twin screws, turning outwards, as they are usually made to do, in order to avoid the risk of floating objects becoming jammed between the upper blade and the ship's counter, should cause least vibration when the shafts are slightly inclined upwards and outwards, starting from the stern; those, on the contrary, which turn inwards would be most favourably placed with the shafts inclined downwards and inwards. Mr. Thornycroft was the first to draw attention to the injurious effect

of an excessive vertical inclination of shaft, and also to the possibility of neutralising the action of the wake by inclining the shaft horizontally in single-screw vessels.

So far as efficiency goes, there seems to be little to choose between outward and inward turning screws.*

The engine-room can sometimes be more conveniently arranged if the screws turn inwards.

Racing.

The racing of screws is due to either of two causes. If the propeller breaks the surface of the water as the stern rises in a seaway, it will draw down air, and the resistance will be immediately very much reduced. Referring to Plate I., where the thrust is shown at different revolutions of a propeller, both when completely immersed and also when splashing, it will be seen that in the former condition a thrust of 11 lbs. is exerted at 680 revolutions.

When air is drawn down, the same thrust is exerted at 1000 revolutions, so that this propeller, if delivering a constant thrust, would vary its revolutions very rapidly from 680 to 1000 if alternately raised and lowered as in the action of pitching.

* See paper by Mr. R. E. Froude in Proc. Inst. Naval Architects, 1898.

But it is not only when the screw breaks the surface that it will race. If a vessel is among waves, racing may occur, although the screw may not be emerged at all. Mr. Froude pointed out that this was probably due to the circular motion of the particles of water in waves. There is no real motion of translation in waves, the water which is travelling in one direction at the crest, returns in the opposite direction in the trough. This circular motion extends to some distance below the surface, and a screw finds the resistance of the water augmented or reduced, as it is beneath the trough or crest of a wave, and reduces or increases its speed accordingly.

Effect on Steering.

A screw causes lateral motion of the stern of a vessel which has to be counteracted by the rudder. This effect is very much greater when going astern than when going ahead, but the cause is not the same in the two cases.

Professor Osborn Reynolds, who has carefully studied this question, attributes the effect of a screw in causing a vessel to diverge from a straight course when going ahead, to the difference in the onward motion of the frictional wake at the surface and at the keel. He agrees with Rankine that the mean speed of the wake of a fairly fine vessel may be 10 per cent. of the vessel's speed,

but thinks it varies from 20 per cent. at the surface to nil at the keel. The upper blade therefore experiences more resistance than the lower, and tends to drive the stern round. If the screw is right-handed and does not draw down air, it will tend to cause the vessel to carry a starboard helm in order to maintain a course. If there is air in the wake, caused, for example, by the vessel being at a light draught of water, the effect is reversed, the lower blade predominates, and port helm must be carried. The natural effect of the screw may also be neutralised or even reversed if there is a broad counter over it and a large rudder, especially if, as is often the case, the part of the rudder behind the upper blade of the screw is larger in area than the part behind the lower blade. The reaction of the stream of water thrown from the upper blade upon the counter and upper portion of the rudder is greater than that of the stream thrown from the lower blade in the contrary direction upon the lower portion of the rudder, and may necessitate the carrying of a port helm with a right-handed propeller.

Professor Reynolds states that a right-handed screw without air always bears considerably on the port side of the stern post, even when the ship carries a port helm, in which case it is obvious that the excess pressure upon the upper part of the rudder must more than counterbalance the excess pressure upon the upper screw blade.

When the screw is reversed and the vessel has gathered stern way, the propeller has a much greater influence upon the course of the ship than when going ahead. In the latter case the influence is always very small, in the former it is often great. The engines will be observed to have a tendency to race when going astern. The screw is then drawing air, and the upper blades being most affected thereby, the lower blades experience the greatest resistance and drive the stern round. A right-handed screw tends to move the stern to port, and a left-handed one to starboard.

When a screw is suddenly reversed, and before the headway is off the vessel, the action of the rudder cannot be depended upon. The following is an extract from the report of the Committee appointed by the British Association to investigate the effect of propellers on the steering of vessels.

British Association Report, 1878.

"It is found an invariable rule that during the interval in which a ship is stopping herself by the reversal of her screw, the rudder produces none of its usual effect to turn the ship, but that under these circumstances the effect of the rudder, such as it is, is to turn the ship in an opposite direction from that in which she would turn if the screw were going ahead. The magnitude of this effect is always feeble, and is different for different ships, and even for the same ship under different conditions of loading. It also appears that owing to the feeble influence of the rudder over the ship during the interval in which she is stopping, she is then at the mercy of any other influences that may act upon her. Thus, the wind, which always exerts an influence to turn the stem of the ship into the wind,

but which influence is usually well under control of the rudder, may, when the screw is reversed, become paramount, and cause the ship to turn in a direction the very opposite of that which is desired.

"Also the reversed screw will exercise an influence, which increases as the ship's way is diminished, to turn the ship to starboard or port, according as it is right or left-handed, this being particularly the case when the ship is in light draught. These several influences—the reversed effect of the rudder, the effect of the wind, and the action of the screw—will determine the course the ship takes during the interval of stopping.

"They may balance, in which case the ship will go straight on, or any one of the three may predominate."

Notwithstanding that a screw has a tendency, as just described, to produce sideways motion of the stern, and so to cause a vessel to deviate from a straight course, it yet offers considerable resistance to lateral motion produced by external causes. The pressure on the blade which is moving in the same direction as that in which the stern of the vessel is turning is increased, while on the blade moving in the opposite direction, the pressure is reduced, that is to say, if the screw is right-handed and the vessel is under port helm, the stern consequently travelling to port, the resistance of the lower blade, which is moving towards the port side, will be increased, and the resistance of the upper blade, which will be moving towards the starboard side, will be diminished, because the one is meeting the water and the other is receding from it. The change of pressure will be proportional to the square of the angular velocity of the stern. The

irregular pressure causes the vibration frequently noticed when a screw vessel is rapidly turning. This resistance to lateral motion is not without value, because if it is removed the condition of a vessel moving in a straight line is one of instability. If the vessel makes the least angle to the direction in which she is moving, the excess of pressure due to undisturbed water at the bow tends to increase the divergence, and this tendency is resisted by the propeller.

A Steering Screw.

A very ingenious propeller has been proposed which, while retaining the advantage of rigid blades in resisting lateral motion when a vessel is on a straight course, is so arranged that the blades are under control and can be feathered in such a way as to cause them not only to offer no resistance to turning, but to actively assist in it. It is in fact a steering propeller of a very simple description. It is illustrated in Figs. 24, 25, 26.

The screw shaft terminates in a universal joint which connects it with an extension of itself in the form of a smaller shaft or tail-piece; the joint can be made in several forms, but the simplest is shown in Fig. 24, where the centre or connecting piece has four arms at right angles to each other. The blades are rigidly fixed to the two arms, which are directly held by the jaws of the main shaft. See Fig. 24, which shows the screw in elevation.

Any movement of the tail-piece causes an alteration in the pitch of the blades, an increase in the pitch of one blade being accompanied by an equal decrease in the pitch of the other. The effect of the change of pitch is that the stern of the vessel is forced either in one direction or the other, according to which side the tail-piece is moved. When it is desired to change the course of the vessel to the right, the tail-piece is moved to the right, its manipulation being thus similar to that of the rudder to which it may be attached, as shown in Figs. 25 and 26. The joint is covered with a light casing, as shown in Fig. 25. The screw may have two, three or four blades. A characteristic which adds much to the practical value of the design is that the feathering of the blades is greatly assisted by the action of the water-flow itself, as any alteration in the course of the vessel tends to change the pitch of the blades in such a manner as to bring the tail-piece into that position which would of itself cause such an alteration, so that after having initiated the feathering

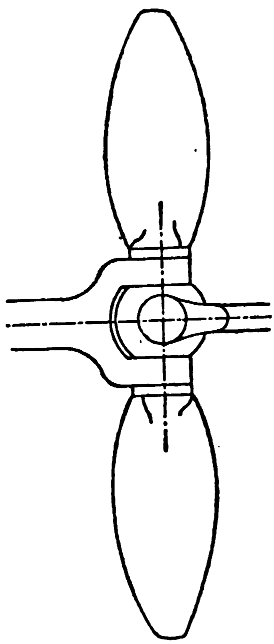


FIG. 24.

motion, it may be anticipated that the tail-piece will have little more to do than to control and regulate it. It is the patent of Mr. F. H. White.

When auxiliary steam power is to be applied

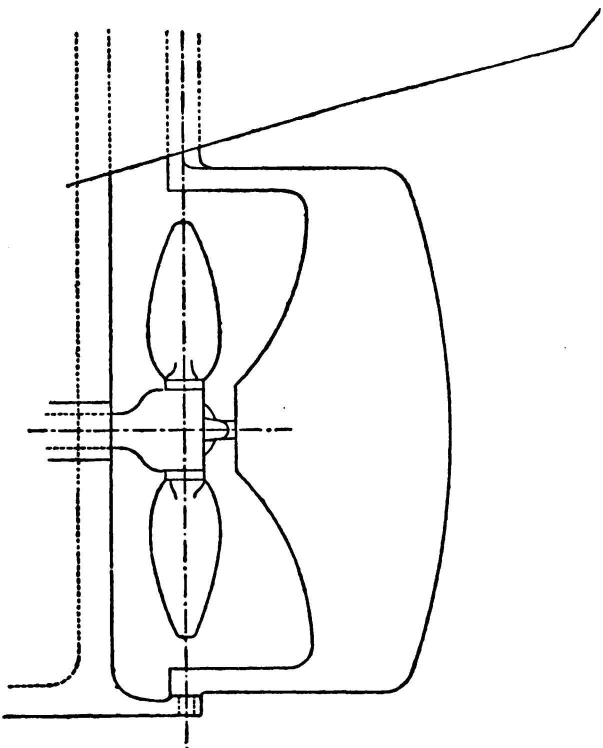


FIG. 25.

to vessels, it is desirable, if possible, to arrange the screw so that it can be lifted out of the water. If disconnected and allowed to revolve, it causes considerable resistance. In cases where it cannot

conveniently be lifted, it is advisable to use a screw with two narrow blades, which can be set up and down in a line with the stern post, and feathered

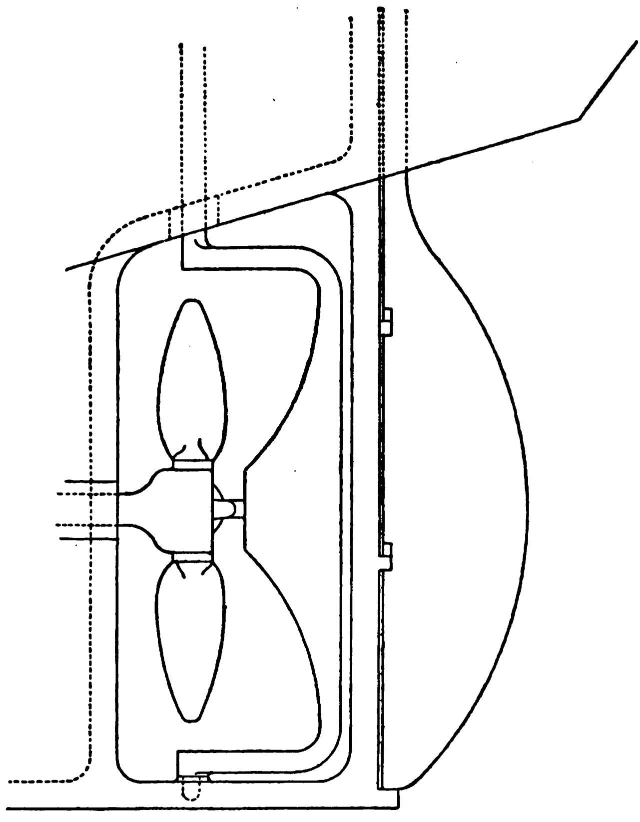


FIG. 26.

fore and aft by internal mechanism as arranged by Bevis.

With a view of reducing the drag under sail to a minimum, Messrs. Thornycroft tried a pro-

peller with flat blades, which could be feathered in line with the shaft for sailing, or set at an angle with it for steaming, but it was found to be a very inefficient propeller, requiring double

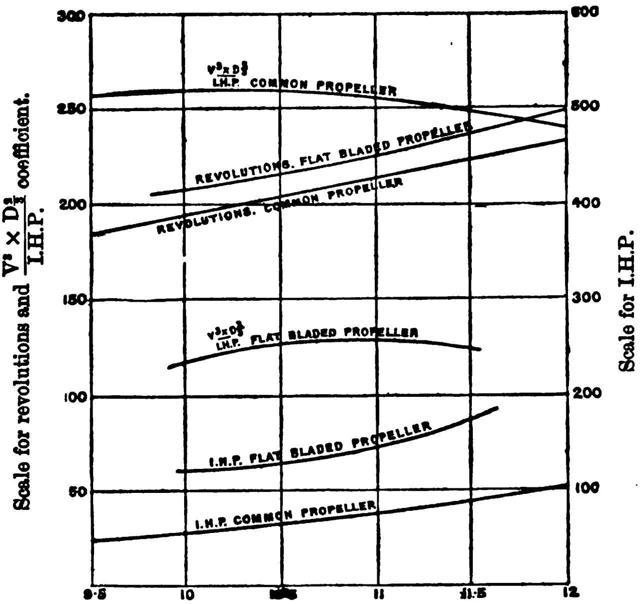


FIG. 27.—SPEED IN KNOTS.

Comparative trial with common and with flat-bladed propellers.

as much horse-power for a given speed as was needed with an ordinary screw.

Fig. 27 shows the results obtained with it, compared with those given by an ordinary screw by which it was ultimately replaced.

Screws for Tugs.

Vessels intended for towing require large screws, because if the screws are designed to work at their best efficiency against the small resistance of the tug running alone, the slip when towing will be excessive, and will cause an undue waste of power. It is desirable to so design the propeller that it shall have a good efficiency at the speed which the tug may be expected to attain when towing an average load.

Screws for Electric Launches.

Screws for electric launches labour under the disadvantage of having to run at an exceptionally high rate of revolution. The blades should be very thin, sharp, and well polished, and two blades will give less resistance to turning with a given surface than three. It is difficult to estimate the pitch necessary to give a required number of revolutions, because it is a peculiarity of the motor that the slower the rate at which the screw turns the faster the power is run down, and *vice versâ*. It may be compared, in this respect, to the steam siren, which uses a large amount of steam when revolving slowly, and the more rapid the rate of turning the less is the quantity of steam passed.

The fastest running screws of which the author

has had experience were designed by him for the Howell torpedo. They were 6 inches in diameter, and ran at the rate of 5000 revolutions per minute, driving the torpedo at 30 knots.

Twin Screws.

Twin screws possess many advantages over a single screw, and are quite as efficient. In order to reduce the lengths of outside shafting in very

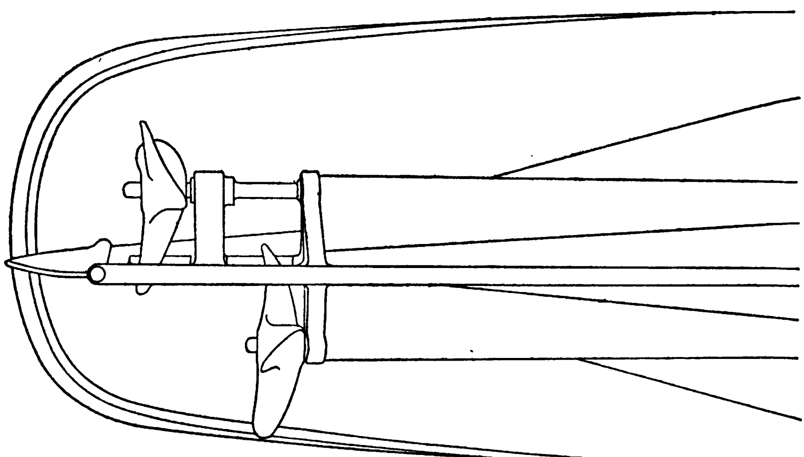


Fig. 28.

fine ships, the discs of the screws are sometimes made to overlap, as shown in Figs. 28 and 29, the blade tips passing through an aperture in the dead wood. Among notable examples of vessels with this arrangement of screws may be mentioned the *Teutonic* and *Majestic*, built by Messrs. Harland and Wolff.

The screws of the *Teutonic* are 19 feet 6 inches in diameter, and the distance between the shafts is 16 feet. One screw is 6 feet 3 inches behind the other. They are right and left-handed, and turn outwards. In a twin-screw torpedo boat built by M. Normand, with overlapping screws, both are arranged to turn the same way. The

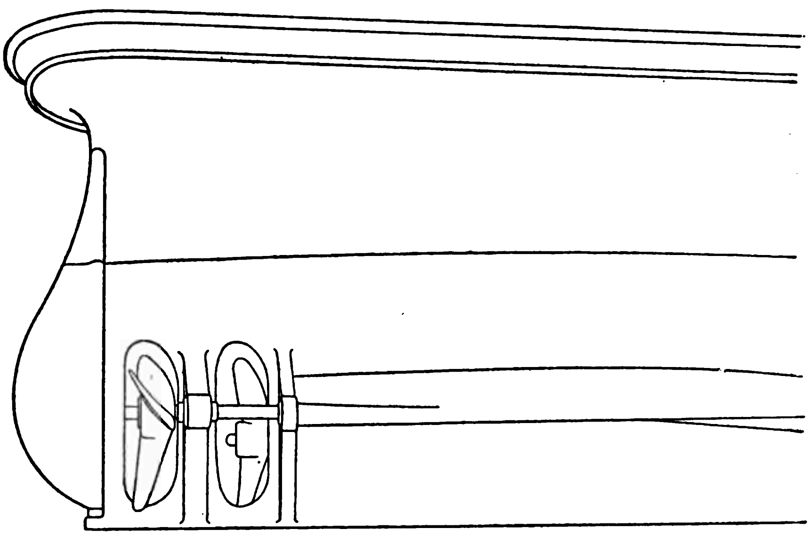
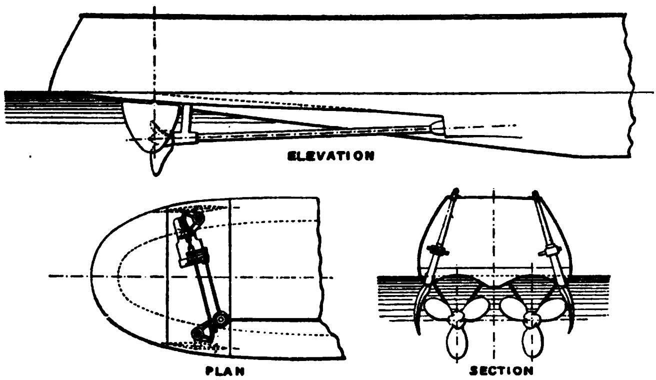


Fig. 29.

overlapping blades thus cross one another, and the slip is reduced. It is found that, when so arranged, the aftermost propeller turns slower than the foremost one, the contrary being of course the case when they turn in opposite directions.

An arrangement of twin screws introduced by

Messrs. Thornycroft, and shown in Figs. 30 to 31A, has proved to be very efficient in small vessels. There is an unobstructed flow of water to the screws, and the concave form of the hull over them serves to collect and convey to them the water which has been put in motion by the friction of the vessel, thus reducing their slip (see page 45). There is but little pitching motion of



FIGS. 30, 31, 31A.

the stern in a seaway, its great width causing it to follow closely the movements of the water surface.

The screws can be placed nearer the surface than would be prudent if the stern were of the usual form, as there is less tendency to race, and they are so well covered by the hull that air cannot get to them.

It is important that the length of the outboard portion of the shaft should be kept within such limits as will preclude all possibility of "whirling"

motion being set up in it so long as it is supported by the shaft brackets. The following formula for this length is given by Rankine.*

$$\text{Maximum length of shaft between supports} = \pi\left(\frac{E\,g\,r^2}{w\,a^2}\right)^{\frac{1}{4}}.$$

Where $\frac{E}{w}$ = modulus of elasticity,

r^2 = square of radius of gyration of cross section,

a = angular velocity of rotation,

= $2\pi \times$ revs. per second.

Taking a modulus of elasticity of 7,500,000 the span in feet becomes for solid shafts

$$176 \cdot 9\sqrt{\frac{\text{diam. in inches}}{\text{revs. per min.}}},$$

and for hollow shafts,

$$176 \cdot 9\sqrt[4]{\frac{D^2 + d^2}{\text{revs.}^2}}.$$

In cases where shaft-brackets have been carried away in torpedo boat destroyers the accident has probably been caused by the breaking of a propeller blade. The commotion at the stern following the loss of a screw blade at full speed in these vessels is appalling, and very great stresses are thrown upon the brackets. The excentricity of the weight and thrust of the unbalanced screw produces a rimering action upon the bearing, and also tends to bend the shaft and induce "whirling," and this tendency to "whirling" is aggravated by the speed at which the engine will run away.

It is prudent to keep well within the limits of unsupported length given by the above formula.

* 'Machinery and Millwork,' p. 550, and *ante*.

TRIPLE SCREWS.

Triple screws are rapidly advancing in favour for ships of war. They appear to possess several advantages. A triple-screw ship has a better chance than a twin-screw ship of escaping total disablement by shot. The engines can be made lighter, because they can be designed to run faster. They can be more easily placed under a protective deck, because they are smaller. They cause less vibration, because the weights of reciprocating parts are less, and because three engines are less likely to synchronise than two.

Admiral Melville, Engineer-in-Chief of the United States Navy, states * that experience with three-screw ships in the American Navy has shown a gain in efficiency of propulsion as compared with twin screws, which he estimates at about 5 per cent. for 15-knot ships, and 12 per cent. for 24-knot ships. He attributes the improved performance to the fact that the centre screw works in the frictional wake, while with twin screws, as usually arranged, the greater part of the wake passes between them. It has been explained (page 66) that when twin screws are placed under a stern shaped as shown in Fig. 31A full advantage of the frictional wake is taken, but it is perhaps questionable if this form would be suitable

* Proceedings of the Inst. of Naval Architects, 1899.

for application to a ship of very large size, and there seems some justification, therefore, in the contention that triple screws should be slightly more efficient than twin screws associated with a stern of the usual shape. The reasoning is, however, open to one objection, which is, that it would lead to the conclusion that twin screws should be less efficient than single screws, and this does not seem borne out by facts.

Moreover, an increase of 12 per cent. in propulsive efficiency seems more than could be expected to result from a mere reduction of the slip of one of the screws, for it is only the centre screw which can be beneficially affected, the wing screws being necessarily as much, if not more, out of the influence of the frictional wake than twin screws would be. More experience is necessary before it can be established that the propulsive efficiency is higher with triple screws.

Admiral Melville recommends that the centre engine should be designed to give approximately one half the power, the remaining half being divided between the two wing engines (see Fig. 32). He claims with justice for this arrangement many advantages. One is that it fits the ship better, there being always less height available in the wings than at the centre line, because of the rise of the bilge and the curvature of the armoured deck. Another is that the smaller diameter of the wing screws tends to shorten the length of outside shaft-

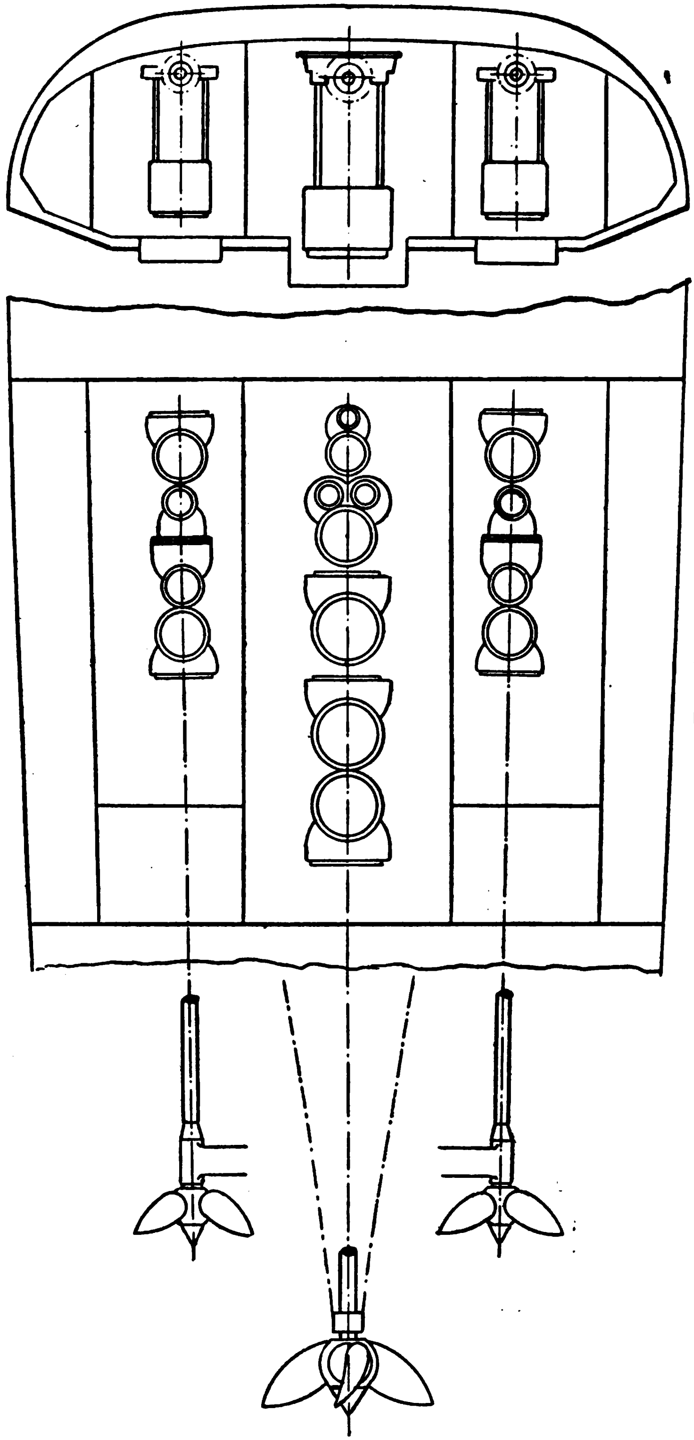


Fig. 32.

ing, and to lessen the projection of the propeller blades; it also enables them to be more deeply immersed, and reduces the chances of their racing as the ship rolls in a seaway.

For cruising purposes the centre screw is disconnected and both wing screws used, the economy of propulsion being considerable. Twin-screw ships when cruising at low speeds, steam with both engines in preference to dragging a disconnected screw and using helm to keep straight.* The engines are consequently worked in an uneconomical manner, the losses from condensation being large. But with the proposed arrangement, by which one-half the total engine power only is employed, there is less engine friction and better steam economy, the engines working at a greater proportion of their maximum power.

Admiral Melville says that in a ship of 12,000 tons, 22 knots, and 23,000 I.H.P., having three equal screws, the power absorbed in dragging one disconnected screw amounts to 150 I.H.P. at 10 knots and to 600 I.H.P. at 15 knots.

If coupled to the engine 300 I.H.P. is required to drag it at 10 knots.

When the middle screw is designed for half the power, still more power must be absorbed in

* Admiral Melville states that $6\frac{1}{2}°$ to $10°$ of helm are needed to keep a course. In ships of very narrow beam in proportion to length, like torpedo boats, it is sometimes more economical to disconnect one screw.

dragging it, not only because of its greater size, but also because it will be more difficult to turn on account of its pitch-ratio being less, so that in such a ship as mentioned above the wasted power would considerably exceed 600 at 15 knots.

It seems to be worth while to consider whether it would not be found more economical to use one of the auxiliary engines, say a circulating pump engine, made especially large for the purpose, to turn the centre propeller at a rate of revolution corresponding to the speed of the ship, since it should take considerably less than 600 I.H.P. at 15 knots, and less than 150 I.H.P. at 10 knots, to overcome the friction of the disconnected shaft and propeller. A circulating pump engine is suggested, because it can be conveniently placed near the shaft coupling. The idle circulating engine might be arranged to drive either the shafting or the pump as required, but it would seem preferable to make one of the wing pumping engines large enough to do this in addition to its pumping work, and avoid introducing the friction and heat wastes of another engine.

CHAPTER IV.

EXPERIMENTS WITH MODELS, AND THEIR APPLICATION TO THE DETERMINATION OF THE MOST SUITABLE DIMENSIONS.

For a screw of any given pitch-ratio there is a particular slip-ratio corresponding to its maximum efficiency. A greater or lesser amount of slip than this will result in a smaller return of useful work in proportion to the power expended in driving the screw. By slip-ratio is meant the ratio P R to V, where

P = mean pitch;
R = revolutions;
V = velocity of feed, or speed of screw through the water.

It has been ascertained experimentally that in the case of a screw of a particular standard form the real slip, as measured by pitch multiplied by revolutions, at different pitch-ratios, should be of the amount given in the following table (see next page) in order to obtain the best results so far as screw efficiency is concerned.

The experiments by which these results were determined were made upon a series of model

screws of a selected type, each differing from the other only in the ratio of pitch to diameter.

TABLE I.

Pitch-ratio.	Real Slip of Screw.	Pitch-ratio.	Real Slip of Screw.
·8	15·55	1·7	21·3
·9	16·22	1·8	21·8
1·0	16·88	1·9	22·4
1·1	17·55	2·0	22·9
1·2	18·2	2·1	23·5
1·3	18·8	2·2	24·0
1·4	19·5	2·3	24·5
1·5	20·1	2·4	25·0
1·6	20·7	2·5	25·4

The following conditions may be laid down as essential if the information obtained from the trials of model screws is to be useful for general application.

1. Each model must be tried at a number of different slip ratios.

2. The velocity of feed must be capable of accurate measurement.

3. The power expended in driving the screw must be measured, and it must be the power put into the screw shaft and not complicated with engine friction, which is an unknown quantity.

It follows that no experiments would be satisfactory in which the screw under examination was

working in the wake of a vessel, because it would then be impossible to measure the velocity of feed, since the forward motion of the wake is an unknown quantity, and varies with the speed of the ship in an unknown manner.

It would be possible thus to ascertain the most suitable propeller for the vessel upon which the experiment was carried out, or for a vessel of similar form with screws similarly situated. It would indeed be the best way of doing so, full size screws being used, but it would be difficult to apply such information as was obtained to the purpose of designing a screw for a vessel of a different form.

The late Mr. W. Froude, by means of an ideal conception of a small element of helical surface rotating at the end of a non-resisting radial arm, deduced by theory for the screw results very similar to those which were afterwards yielded by experiment.*

In 1877 Mr. Froude described how such experiments were being conducted at Torquay,† and a very full account of the system pursued, not only for ascertaining the screw efficiency, but also for investigating the effect upon the operation of the screw of the presence in front of it of the hull of the ship, was given by Mr. R. E. Froude in 1883.‡

* Trans. Inst. Naval Architects, xix. p. 47.

† Proc. Inst. Civil Engineers, li. p. 38.

‡ Trans. Inst. Naval Architects, xxiv. p. 231.

All these papers should be consulted by anyone proposing to experiment for himself.

In the years 1879–80, Mr. John I. Thornycroft made a number of experiments with models of small dimensions, and a detailed description of these will be given because they are interesting as showing what can be done with moderately simple appliances.

The models were about 9 inches in diameter, and this size was found to be convenient. The maximum thrusts did not exceed 30 lbs., so that although the scale was large enough to admit of accurate measurement, a moderately small dynamometric apparatus could be employed. They were made as follows:—A wooden block was prepared from the reduced propeller drawing, upon which a blade was modelled by hand in paraffin.

A mould of the blade was made in plaster of Paris, into which was run an alloy consisting of tin and bismuth, the latter in small proportions. This material is sufficiently soft to be scraped and cut with a knife, and at the same time is strong enough to retain its form. It, of course, does not rust. The cast blade was filed, burnished, and accurately adjusted to the wooden block if it had become at all distorted in casting. The blades were secured in the boss by screws, the holes in the flanges being elongated in the manner adopted by Griffiths, so that the pitch could be varied to any desired extent (see Fig. 49). A steam launch

was fitted up with a small shaft passing through the bow to carry the model screw, the shaft projecting a sufficient distance in front of the launch to ensure that the model should work in undisturbed water. This shaft could move very freely in its bearings in a fore and aft direction, and the inboard end of it was attached by means of a steel pianoforte wire to a spring, so that the thrust exerted by the propeller could be recorded.

This shaft was made to revolve by means of a gutband working on to a pulley, and driven by a small engine of one or two horse-power. The following measurements made were—

1. The thrust exerted by the model;
2. The revolutions of the model;
3. The speed of the launch;
4. The turning effort expended in driving the model;
5. Equal intervals of time.

The constant friction of the engine and shafting was also measured in order to obtain the true zero for the turning effort diagram. A dynamometer was constructed by which records were continuously taken upon a revolving drum driven at a uniform speed by means of clockwork. A number of pens over the drum were each connected to an electro-magnet in such a way that so long as no current flowed the pens were stationary, and traced straight lines upon the paper as it revolved beneath them. When contact was made the pens

were jerked, and made a lateral indent in the line. One pen was electrically connected with a clock, and measured intervals of time, making an indent every twelve seconds. A second pen recorded the revolutions of the main engine driving the launch, these revolutions affording a means of checking the speed. A third pen recorded the revolutions of the model, a counter on the shaft making contact every fifty revolutions.

The speed of the launch was measured by passing a fixed distance on shore of 300 feet, the time of passing the posts being marked by one of the electric pens, actuated by the observer pressing a button. As the observations were taken in a tideway, two runs were necessary to determine the speed for every observation, one up stream, and one down.

Another pen was connected to the spring before mentioned, to which the model shaft was attached, and recorded the extension of the spring and thrust of the propeller.

The last pen showed the tension of the gut driving the model, and thus measured the turning effort upon the shaft. This tension was obtained by the arrangement shown in Fig. 33. The large pulley in the centre is driven by the small engine of which the cylinder, piston-rod, connecting-rod, and crank-arm are indicated in the figure. The lower pulley is on the shaft of the model propeller. The two upper pulleys are carried by a bar pivoted

at the centre. Rigidly attached at right angles to this bar is a long lever, the weight of which is balanced by the ball at the top. The motion of the lever is limited to a short travel on each side of the vertical by stops not shown in the figure. A spring is attached to the bottom of the lever,

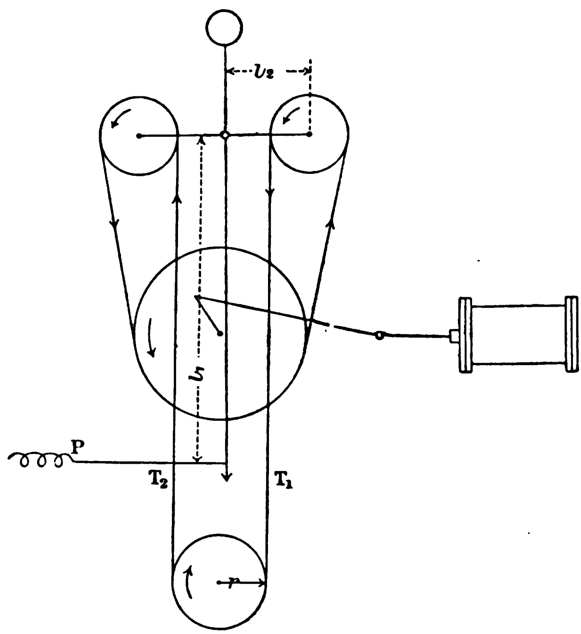


FIG. 33.

and its extension is automatically recorded upon the diagram. The driving band is passed round the pulleys as shown, the direction of its motion being indicated by arrows. When the central pulley is made to revolve, the tension of the gut pulls down the left-hand pulley, and extends the

spring until its tension is sufficient to prevent further motion of the lever. An adjustment is provided in the cord between the spring and the lever so that the latter may be maintained approximately vertical.

If T_2 is the tension of the ascending or tight side of the band, T_1 the tension of the descending or slack side, P the pull in pounds as measured by the spring, and l_1 and l_2 the lengths of the levers as shown, then

$$T_2 - T_1 = \frac{P \times l_1}{2\, l_2},$$

and turning moment $= (T_2 - T_1) \times 2\,\pi r \times$ revolutions of the pulley r per minute.

The launch, which as before stated was driven by an independent screw, maintained an approximately constant speed of about $4\frac{1}{2}$ knots, and a number of observations were taken at different revolutions of the model, and plotted as shown in Plate I. Curve A is the thrust of the model, B is the useful work in foot-pounds per minute, being the product of the thrust into the speed through the water. C is the work expended in foot-pounds per minute. The useful work divided by the work expended is a measure of the efficiency of the model as shown by curve D.

A convenient way of utilising the results thus obtained is to construct a series of constants which will express the relation between disc-area, power, and speed, at different slip-ratios. A second series

of constants can be formed expressing the relation between diameter, speed and revolutions.

These constants depend upon the following laws :—

1. For a given pitch-ratio and efficiency the disc-area is proportional to the horse-power and inversely proportional to the cube of the speed.

2. For a given pitch-ratio and efficiency the revolutions per minute are proportional to the speed and inversely proportional to the diameter.

They might take the following forms :—

$$C_A = \text{disc-area in square feet} \times \frac{v^3}{\text{H.P.}};$$

$$C_R = \text{revolutions per minute} \times \frac{D}{v};$$

when v = velocity of feed, and H.P. = effective horse-power in the screw shaft.

In this shape, however, it would only be possible to obtain directly from them the proportions proper for a screw to propel a "phantom ship," that is a ship which would require the same thrust to propel it at any given speed as a real ship, but which will create no disturbance in the water, driven by a "phantom engine"—that is, an engine without friction in which the brake horse-power is equal to the indicated horse-power. In order to make them available for general use, it is more convenient to substitute V = speed of ship for v = velocity of feed, and I.H.P. for effective horse-power in the shaft. In order to do this it is necessary to

make certain assumptions as to the speed of the following current and as to the ratio $\frac{E.H.P.}{I.H.P.}$, and as these will vary with the form of the ship, and with the type of engine respectively, an element of uncertainty is here introduced, and much will depend upon the judgment of the designer, as to whether it is necessary to apply a wake correction or a propulsive coefficient correction, or whether the standard values assumed for these factors may be supposed to be a sufficiently close approximation.

Let it be supposed that the resistance of a vessel at a given speed is known—say, by measuring the pull in the tow-rope. In order to propel the vessel at this speed by means of a screw at the stern, a thrust will have to be exerted by the screw greater than the pull of the tow rope by an amount varying to some extent with the pitch-ratio and position of the propeller, but which may be assumed for the present purpose at 10 per cent. If the screw works as it generally does in a forward current produced by the friction of the vessel, its speed through the water will be less than the speed of the vessel by an amount varying very much under different circumstances. The velocity of this current will depend, as previously explained, page 45, upon the length of the ship, the form of the lines, the state of the bottom and the position of the propeller. Let it be assumed that it is

10 per cent. of the speed of the vessel; in other words, that the velocity of feed is 10 per cent. less than the ship's speed. Again the indicated horse-power of the engines will exceed that transmitted to the screw by an amount equal to the power expended in internal friction. This ratio may be taken as I.H.P. is to power in shaft as 100 is to 77. Finally taking the mean efficiency of the propeller at 65 per cent. we have all the elements necessary to fix the dimensions of a screw to work behind a ship at the real slip suitable to its pitch-ratio.

Thrust horse-power, or T.H.P. = thrust of screw × velocity of feed.

Effective horse-power, or E.H.P. = tow-rope resistance × speed of ship.

If it be agreed to assume that

$$\text{Tow-rope resistance} = \text{thrust} \times \cdot 9,$$

and

$$\text{Velocity of feed} = \text{speed of ship} \times \cdot 9;$$

then

$$\text{T.H.P.} = \text{E.H.P.}$$

Since I.H.P. is greater than E.H.P. by the amount wasted by the friction of the engine and by the screw, we get

$$\text{E.H.P.} = \text{I.H.P.} \times \cdot 77 \times \cdot 65,$$

$$\therefore \quad \frac{\text{E.H.P.}}{\text{I.H.P.}} = \cdot 5 = \text{propulsive coefficient.}$$

These assumptions were made by Mr. R. E. Froude in his paper upon the determination of

the most suitable dimensions of screw propellers, read before the Institute of Naval Architects in 1886, and the author has adopted them in constructing the constants in the Table II., p. 108. The values both for engine efficiency and for screw efficiency may appear low, but they probably fairly represent the average values obtained in vessels of different classes. Engine efficiencies, by which is meant the ratio $\frac{\text{brake horse-power}}{\text{indicated horse-power}}$, may be as high in some cases as ·85, and propeller efficiencies as high as ·7, in which case the propulsive coefficient would be ·6.

This value of propulsive coefficient is frequently met with in fast steam launches, torpedo boats and destroyers. There have been cases in which it has apparently been considerably exceeded, propulsive coefficients even as high as ·7 having been recorded, but they are quite abnormal and will be dealt with later (see page 100).

The values of C_A and C_B in Table II. are obtained from model trial results at any slip-ratio as follows:—

Let the thrust of the screw in lbs. $= T$
„ velocity of feed in knots $= v$
„ speed of ship in knots $= \left(v \times \frac{1}{\cdot 9}\right) = V$
„ revolutions per minute $= R$
„ disc-area of model in square feet $= A$
„ diameter of model in feet $= D$

And if I.H.P. be assumed $= 2 \times$ E.H.P.

$$\frac{T v}{33{,}000} \times 2 \times \frac{6080}{60} = \text{I.H.P.}$$

Then

$$C_A = \frac{A \times V^3}{\text{I.H.P.}},$$

$$C_B = \frac{R \times D}{V}.$$

Allowance can be made for different values of wake percentage by multiplying the speed of ship V by a wake-correction factor. The standard wake has been taken as 10 per cent. of the speed of the vessel. In a very full ship it might be as high as 30 per cent.; therefore the speed of the ship V should be reduced when using the constants by 20 per cent., in the case of a very full ship whose wake was estimated at 30 per cent.; and by amounts varying from 20 per cent. to nil, as the fulness of form varies from "very full" down to what may be considered as a "fairly fine" form when no correction need be made. If, on the other hand, it is thought that the wake is likely to be less than 10 per cent., the speed of the vessel V must be increased by an amount varying from nil to 10 per cent., the latter correction corresponding of course to "no wake."

Table III., p. 108, gives the value of the wake correction for a few vessels. The standard ratio of $\frac{\text{E.H.P.}}{\text{I.H.P.}}$ has been taken as ·5 as already stated. A correction can be made for any deviation from

this assumed value. If for example the E.H.P. is estimated to be 55 per cent. of the I.H.P., then the I.H.P. must be multiplied by the ratio $\frac{\cdot 55}{\cdot 50}$.

Again, the constants are primarily correct for four-bladed screws. They can be used for three-bladed or two-bladed screws, by multiplying the I.H.P. by $\frac{1}{\cdot 865}$ or $\frac{1}{\cdot 65}$ respectively.

The manner in which the table has been constructed will perhaps be better understood by reference to Plate I.

The constants C_A and C_B corresponding to 750 revolutions of the three-bladed model, whose performance is shown in the curves, would be found as follows :—

$$\text{Diameter of model} = 0\cdot 75 \text{ feet,}$$

$$\therefore \text{ Disc-area} = 0\cdot 441 \text{ square feet.}$$

$$\text{Useful work in foot-pounds} = 6330.$$

$$\text{I.H.P.} = \frac{6330 \times 2}{33{,}000} = 0\cdot 384.$$

$$\frac{\text{Useful work}}{\text{Thrust in pounds}} = \frac{6330}{14\cdot 8} = 426 = \text{velocity of feed in feet per minute.}$$

$$\frac{426 \times 60}{6080} = 4\cdot 22 \text{ knots.}$$

$$C_A = \frac{\cdot 441 \times \left(4\cdot 22 \times \frac{1}{\cdot 9}\right)^3}{\cdot 384 \times \cdot 865} = 137.$$

$$C_B = \frac{750 \times \cdot 75}{4\cdot 22 \times \frac{1}{\cdot 9}} = 120.$$

If from the curves in Plate I. constants are calculated in this way corresponding to different amounts of slip, a series of values of C_A and C_B is obtained such as is shown by any one of the horizontal lines in Table II., and these can be placed in their proper relative positions under a curve of efficiency.

The object Mr. Thornycroft had in view in making his experiments, was to test the efficiency of the screw-turbine propeller as compared with the screw he was using on torpedo boats,* and he did not carry out a complete set of experiments upon a common screw, which would have involved a series of trials with a number of models of varying pitch-ratio, but he contented himself with ascertaining that a small change of pitch on either side of that for which the propeller was cast, obtained by twisting the blades in the boss, did not increase the efficiency but rather reduced it.

This complete series was however afterwards carried out by Mr. R. E. Froude at Torquay, and the results were given by him in a paper read before the Institution of Naval Architects in 1886.† They corroborated generally those obtained at Chiswick, except in one particular.

So far as could be judged by Mr. Froude, there was scarcely any difference in maximum efficiency

* See a paper by Mr. Thornycroft in Trans. Inst. Naval Architects, xxiv. p. 42.

† Idem, xxvii. p. 250.

within such a large range of pitch-ratio as from 1·2 to 2·2. As almost precisely the same maximum was obtained by Mr. Thornycroft with a pitch-ratio of 1·14, it seems reasonable to suppose that these even may hardly be considered as hard and fast lines beyond which efficiency will decline, and Mr. Froude considers that they may be fairly extended to 0·8 on the one side and 2·5 on the other.

It must, however, be borne in mind that this equality of efficiency is manifested by screws working in open water.

There is a very general consensus of opinion that small pitch-ratios give the most favourable results in practice. If this opinion is justified, the explanation must be that large pitch-ratios cause a greater augmentation of hull resistance. That this would be so might be inferred from the reasoning on page 31, which leads to the conclusion that such screws produce a greater suction than those of fine pitch. Even supposing that the parity of efficiency of different pitch-ratios within the limits experimented on were accepted when the screws are considered apart from the vessels they propel, there seems reason to suppose that there will be a loss involved by the use of a screw of very coarse pitch, if it be placed in such a position that the increased suction produced by it is able to take effect upon the hull of the vessel.

Mr. Froude also found a great similarity be-

tween the curves of efficiency at different pitch-ratios, the only apparent effect of change of pitch-ratio being to cause the maximum efficiency to occur at different slip-ratios. Where, for example, the efficiency of one propeller reached a maximum at 15 per cent. slip, the efficiency of another of different pitch-ratio was at a maximum at 20 per cent. slip, and so on (see Table I.). It was therefore possible to superimpose the curves, and cause them to coincide, by the simple device of empirically changing the scale of slip-ratio. Mr. Froude very kindly gave the author permission to give the results of the Torquay experiments in a paper for the Institution of Civil Engineers on "The Screw Propeller," * for which purpose the author compiled Table II., page 108, in which each horizontal line of figures corresponds to a particular pitch-ratio, and contains constants for disc-area, and revolutions at different amounts of slip, calculated from a set of curves such as that shown in Plate I., and in the manner already described. These all occupy their proper relative positions under the curve of efficiency.

The table embraces the whole of the experiments possible with a particular type of screw, including pitch-ratios extending from 0·8 to 2·5, and slip-ratios from the lowest to the highest which is considered practicable.

It would be used in the following manner.

* Proc. Inst. Civil Engineers, cii. p. 74.

Let it be supposed, for example, that the size of the screw is limited by the draught of water. If the given disc-area is multiplied by the cube of the speed of the vessel in knots, and divided by the I.H.P., the constant C_A is obtained. Suppose it is 360. The nearest figure to this in the column under the maximum efficiency should be sought. Its position, when found, will indicate the pitch-ratio, which will be in the same line at the left hand of the table. In the case supposed the pitch-ratio will be 1·6.

Adjoining the disc-area constant 360 will be found the revolution's constant, $C_R = 71$. This number, multiplied by the speed of the vessel in knots, and divided by the diameter of the screw in feet, will give the number of revolutions at which a four-bladed screw should run to obtain the maximum efficiency.

The figures printed in small type between the columns of C_A and C_R constants show the value of $0 \cdot 7854 \frac{C_R^2}{C_A}$, and have been added at the suggestion of Mr. C. H. Wingfield * to simplify the procedure when it is required to design a screw to run at a fixed number of revolutions.

If the value of $\frac{\text{I.H.P.} \times \text{revs.}^2}{V^5}$ be calculated, the nearest number to the result will be found to lie between the constants for diameter and revo-

* Proc. Inst. Civil Engineers, cii. p. 101.

lutions suitable for the prescribed conditions (see Example 4, page 98).

It is evidently desirable to select the constants from the column under the maximum efficiency but if the revolutions are required to be either higher or lower than the number most suitable for the propeller in order to suit the engines, the same disc-area constant may be taken from one of the other columns, where it will be found associated with either a lower or a higher value of C_R, accordingly as the slip-ratio is greater or less; and it is possible to see at a glance what sacrifice it is necessary to make in efficiency in order to attain the required result.

Mr. Wingfield has pointed out that if the product of the C_R constant multiplied by its proper pitch-ratio is greater than $101 \cdot \dot{3}\dot{3}$, the apparent slip will be positive; if less, it will be negative. The amount of the slip in either case will be given by

$$\text{Slip per cent.} = \frac{p\,C_R - 101 \cdot \dot{3}\dot{3}}{p\,C_R} \times 100,$$

where p = pitch-ratio.

The same constants are presented in a graphic form in Plate II., in which each vertical column of Table II. is plotted as a curve, and values of C_A and C_R, corresponding to intermediate pitch-ratios and slip-ratios, may be obtained.

The method of correcting for two-bladed screws and three-bladed screws, and also for different values of wake, is due to Mr. R. E. Froude,

and Table III. was given by him for the purpose of assisting in the selection of a suitable wake correction in his masterly paper already referred to,* which is worthy of the most careful study. The wake value of the *Daring*, estimated from the results of the trials, has been added. Mr. Froude's screws were of uniform pitch, and the blades were elliptical. The width in the middle of the developed blade was $0 \cdot 4 \frac{\text{diameter}}{2}$. It follows that the developed surface, supposing each blade to be a complete ellipse, would be

For a four-bladed screw	developed	surface	=	disc-area	$\times$ 0·4 †
„ three-bladed	„	„	=	„	$\times$ 0·3
„ two-bladed	„	„	=	„	$\times$ 0·2

As the developed area is usually taken as exclusive of the boss, the portions of the ellipse cut off by it must be deducted. A few examples of the use of the tables will be given at the end of the chapter.

It will be noticed that the disc-area constants in the columns under maximum efficiency permit great latitude in the choice of diameter for a given I.H.P. and speed if the efficiency of the screws only is considered apart from the vessels they are designed to propel, and the service these vessels are intended to perform. For example, take the case of a vessel of good form having engines of 500 I.H.P., and expected to attain a speed of 10

* Trans. Inst. Naval Architects, xxvii. p. 250.

† See p. 145 for deduction to be made for area of boss.

knots. From Table II. equal efficiency may be expected with a screw having a diameter of 10 feet and 0·8 pitch-ratio, and with one having a diameter of 15½ feet and a pitch-ratio of 2·5. The first would run at 138 revolutions per minute, the second at 33½. Some remarks have already been made (see page 88) touching the effect of pitch-ratio upon efficiency, but in considering the relative advantages of large and small screws, the duties required of the vessel must be taken into account. Suppose it is desired, for example, to maintain a high speed against head-winds and seas. It is generally supposed that a large screw or a large surface is all that is required, but if we consider what happens when a vessel meets head-winds, we shall see that this is not necessarily so. In such circumstances the speed of the ship is checked, while the revolutions of the screw remain practically unchanged, so that the slip-ratio is increased. If this is already sufficient to give maximum efficiency at the smooth water speed, the efficiency will be reduced when the slip is increased by the wind; and it is probable that the 15½-foot screw of 2½ pitch-ratio would waste as much power as the 10-foot screw of 0·8 pitch-ratio, although one has nearly 2½ times the surface of the other, because they both have the same position to start with as regards the curve of efficiency. Large diameter associated with large pitch-ratio is valueless for the purpose; what is required is, that the

slip-ratio shall not be excessive when the speed of the vessel is retarded by external resistances. The case is analogous to that of a tug, and must be similarly treated. The best proportions will be obtained by designing for a speed less than the maximum smooth-water speed, but such as the vessel is expected to maintain over an average passage. The propeller would have rather less than the maximum efficiency when the vessel was developing her full power over the measured mile —would, in fact, be too large—but would work at its best at the speed assumed as the average, and should effect a saving of fuel on the voyage.

It has been stated by Mr. Hall-Brown * that for vessels of very full form—a class with which he has had much experience—a large diameter and a small pitch-ratio are essential to success, and he attributes the necessity to the influence of dead-water as distinguished from frictional wake (see page 46). In such a case the blades must reach well out into water clear of the stern, so that the proportion of dead water to the total area of stream acted upon may be as small as possible. Mr. Hall-Brown gives the particulars of what is found to be a good screw for a cargo vessel of the following dimensions :—

Length B.P.	277 feet
Beam moulded	37·5 ,,
Draught	19 feet 11 inches

* Proc Inst. Civil Engineers, cii. p. 131.

Displacement	4670 tons
Block coefficient	·792
I.H.P.	825
Speed	9 knots
Diameter of screw	16 feet
Pitch of screw	16 „
Revolutions of screw	64

The values of C_A and C_R are 184 and 115 respectively, which agree with the constants in the table for 1·0 pitch-ratio at 69 per cent. efficiency, but as the table is calculated for a wake value of 10 per cent. corresponding to a fine form of vessel, while the cargo steamer has a full form, and a much greater wake percentage, this close agreement with the table would not have been expected. He suggests that the explanation is that the propulsive coefficient is very low on account of the action of the screw upon the dead-water, and the two corrections for wake and dead-water tend to annul one another.

In such a case it might be expected that a better performance would be obtained if twin screws were employed, since these would be to a greater extent clear of the dead-water.

Whenever the ship is of exceptional form, no exact rules can be given for the proportions of screws deduced from model trials in undisturbed water. Certainty can only be arrived at by trying the model screw behind a model of the ship, and this is always done by Mr. Froude in the case of new Admiralty designs. But every carefully

recorded ship trial is in one sense a model experiment of this character, and when a screw is to be designed for a ship of a special type, it is safer to calculate the values of C_A and C_B from the actual figures obtained from the trials of some vessel of proportions as similar as possible, which has given a good result, because the values of wake and propulsive coefficient may then be assumed to be similar also. In place of constructing constants, the following more direct method may be employed —

To find the diameter of a propeller for a given I.H.P., and a given speed from the diameter of another similar propeller at a different I.H.P. and a different speed.

If d = diameter of model, which may be larger or smaller than D,
D = diameter of required propeller,
p = I.H.P. of model,
P = I.H.P. of required propeller,
v = speed of vessel with model propeller,
V = speed of vessel with required propeller,
r = revolutions of model propeller,
R = revolutions of required propeller,

then

$$D = \sqrt{d^2 \times \frac{v^3}{V^3} \times \frac{P}{p}},$$

and

$$R = r \times \frac{V}{v} \times \frac{d}{D}.$$

The pitch ratio must be the same as that of the screw, which is treated as the model.

Examples of the Use of Tables II. and III.

Example 1.—Find the diameter and revolutions of a 4-bladed screw to work at maximum efficiency for a vessel of 20 knots speed and 6000 I.H.P. Pitch-ratio to be 1·2.

The disc-area constant (C_A) in Table II. for this pitch-ratio, and in the column over 9 abscissa value is 288, and the revolutions constant (C_R) is 92.

$$\text{Disc-area} = C_A \times \frac{\text{I.H.P.}}{(\text{speed in knots})^3} = 288 \times \frac{6000}{20^3} = 216 \text{ sq. ft.}$$

$$\therefore \text{ Diameter} = 16 \cdot 5 \text{ feet.}$$

$$\text{Revolutions} = C_R \times \frac{\text{speed in knots}}{\text{diameter in feet}} = 92 \times \frac{20}{16 \cdot 5} = 111.$$

Example 2.—Find the pitch and revolutions of a 4-bladed screw to work at maximum efficiency for a vessel of 20 knots speed and 6000 I.H.P. Diameter not to exceed 15·5 feet.

$$\text{Disc-area} = 189 \text{ square feet.}$$

$$C_A = 189 \times \frac{20^3}{6000} = 252.$$

The nearest disc-area constant in the table under maximum efficiency is 251 at pitch-ratio 1·0.

$$\therefore \text{ Pitch} = 15 \cdot 5 \text{ feet.}$$

The corresponding value of C_R is 109,

$$\therefore \text{ Revolutions} = 109 \times \frac{20}{15 \cdot 5} = 141.$$

Example 3.—Find the pitch-ratio and efficiency of a 4-bladed screw for a vessel of 20 knots speed and 6000 I.H.P.

The diameter to be 15·5 feet and the revolutions about 80 per minute.

$$\text{Disc-area} = 189 \text{ feet.}$$

$$C_A = 189 \times \frac{20^3}{6000} = 252.$$

$$C_R = 80 \times \frac{15 \cdot 5}{20} = 62.$$

The nearest constants in the table are at pitch-ratio 2·2, and efficiency 68 per cent. When the diameter and revolutions are both limited, the curves on Plate II. will probably be found more convenient, as intermediate pitch-ratios can be selected.

Example 4.—Find the diameter and pitch of a 4-bladed screw to work at maximum efficiency for a vessel of 20 knots speed and 6000 I.H.P.

Revolutions to be 85 per minute.

Wake correction to be made for a form of the fulness of H.M.S. *Devastation*, corresponding to a wake percentage of 15·6.

The multiplier for wake from Table III. is 0·934.

$$20 \times 0 \cdot 934 = 18 \cdot 8 \text{ knots nearly.}$$

$$\frac{\text{I.H.P.} \times \text{revs.}^2}{V^5} = \frac{6000 \times 85^2}{(18 \cdot 8)^5} = 18 \cdot 5.$$

This number will be found between the constants 306 and 85 for disc-area and revolutions respec-

tively at 1·3 pitch-ratio, and 9 abscissa value, which will give

$$806 \times \frac{6000}{(18 \cdot 8)^3} = 276 \text{ square feet.}$$

$$\therefore \quad D = 18 \cdot 75 \text{ feet,}$$

and

$$85 \times \frac{18 \cdot 8}{18 \cdot 75} = 85 \text{ revolutions nearly.}$$

The number 18·5 or near it will be found at all abscissa values from 5 to 16, and any of the constants between which it lies in those columns would give a screw which would run at the desired number of revolutions, but the abscissa value 9 would be chosen by preference unless it were desired to limit the diameter as much as possible without sacrificing efficiency, in which case a pitch-ratio of 1·55 and an abscissa value of 11 could be selected.

Example 5.—Find the diameter, pitch and revolutions of a three-bladed screw to work at maximum efficiency, for a vessel of 20 knots speed and 6000 I.H.P. Pitch-ratio to be 1·2.

$$C_A = 288. \quad C_R = 92.$$

$$6000 \text{ I.H.P.} \times \frac{1}{0 \cdot 865} = 6940.$$

$$288 \times \frac{6940}{20^3} = 250 \text{ square feet.}$$

$$\therefore \quad D = 17 \cdot 8 \text{ feet.}$$

$$92 \times \frac{20}{17 \cdot 8} = 103 \text{ revolutions.}$$

$$\text{Pitch} = 17 \cdot 8 \times 1 \cdot 2 = 21 \cdot 3 \text{ feet.}$$

Note on Abnormal Propulsive Coefficients.

It was said on p. 84 that propulsive coefficients as high as ·7 have sometimes been recorded.

Such very high coefficients are probably due either to the underlogging of the power of fast-running engines by the indicator, or else to some special conditions attaching to the trial, such as a depth of water at the measured mile bearing a relation to the speed and draught of the vessel, which allowed what is known as a "canal wave" or "wave of translation" to be generated. In a channel of rectangular section, the speed V of this wave of translation is equal to $\sqrt{g\,h,}$ where h = height of crest of wave above the bottom of the channel.

When the channel is not rectangular the wave is still formed, although its speed is not so exactly calculable, and when the speed of the vessel coincides with the speed of the wave, the resistance is less than the resistance in normal conditions, and less than the resistance estimated frcm tank experiments, and the propulsive coefficient calculated from the E.H.P., measured in the tank, would not represent the proper ratio of I.H.P. to E.H.P.

Some interesting experiments have been made in Denmark, upon the effect of different depths of water upon the resistance of a torpedo boat at

different speeds. The results were described by Captain A. Rasmussen, Engineer-in-Chief of the Danish Navy, in a paper communicated to the Institution of Naval Architects,* in 1899.

The dimensions of the boat were:—Length, 145 feet 6 inches; breadth, 15 feet 6 inches;

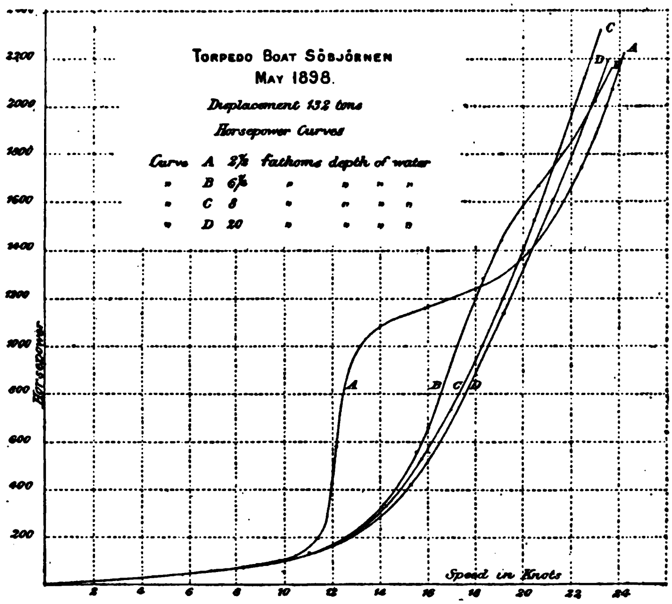


FIG. 34.

draught, fully equipped, forward 3 feet 10 inches, aft 7 feet 9½ inches; corresponding displacement, 140 tons. The trials, the results of which are shown in Fig. 34, were made at a displacement of 132 tons.

The boat had a stern of the Thornycroft form

* Trans. Inst. Naval Architects, vol. xli. p. 12.

(see Figs. 30 and 31A), and the draught aft would be taken from the bottom of the screw. The draught amidships, which is probably the most important as regards wave disturbance, is not given, but would be about 5 feet. Fig. 34 shows the horse-power curves at four different depths of water (2½, 6¼, 8, and 20 fathoms). Captain Rasmussen says, "It will be seen that at half power the loss in speed in shallow water is very great, while at full power the speed is higher for depths below or above 8 fathoms, this being of the four named depths the most disadvantageous for the propulsion of this boat at full power.

"The results as to speed on these trials at about full and half power are as follows:—

Depth of Water.	Speed at 2200 I.H.P.	Speed at 1000 I.H.P.
fathoms	knots	knots
2½	24·1	13·1
6¼	23·8	17·2
8	22·8	18·3
20	23·6	18·6

"The speed corresponding to the points of inflexion in the curves is practically the speed V of the wave of translation, as given by the formula $V = \sqrt{g h}$, when g is the acceleration of gravity and h the depth of water.

"While the wave of replacement in shallow

water at half-power is unusually high and long, it vanishes completely at full speed, the boat is then running on the crest of the wave of translation, the resistance to propulsion being smaller, and the speed therefore higher than in deeper water, where the usual wave system has to be created."

The improvement which takes place in the performance of small vessels when forced to a high speed in deep water is due to a different cause. Not only does the propulsive coefficient reach a higher value, being affected chiefly as already explained by the underlogging of the power with fast-running engines, but the rate at which the resistance of the vessel increases with the speed undergoes a change after a certain critical speed has been passed, and it is this which has made it possible for small vessels to attain to speeds which were formerly thought to be impossible.

There are some who attribute this variation from the law that resistance varies as the square of the speed to a rising of the vessel, due to the vertical component of the direct resistance, and a formula has been given for the speed at which the performance of a vessel is at its worst, and at which the phenomenon of lifting commences to occur. It is $V = K\sqrt{D^{\frac{1}{3}}}$.* The value of the

* See *Engineer*, May 22, 1896.

coefficient K in the formula will depend upon the form of the boat. It has been found that the critical speed at which a change begins to take place in a torpedo boat form is given fairly correctly by the formula if K is taken = 9. Since in similar vessels $D^{\frac{1}{3}}$ is proportional to L, the formula expresses the well-known fact that their corresponding speeds are proportional to the square roots of their lengths. Thus if the torpedo-boat model is supposed to be enlarged first to 300 tons and then to 8000 tons, the critical speeds will be 23·3 knots and 40 knots respectively.

There is no doubt that vessels of suitable form could be made to lift, and more or less to skim on the surface, if sufficient power could be put into them to give them a speed high enough in proportion to their length. A ricochetting shot may be considered to be a vessel having a prodigious speed in proportion to its length. The punt-shaped boats seen on Canadian water-shutes rise nearly out of the water when projected into it with a high velocity, and small models of boats can be towed at a speed which will cause them to skim upon the surface, their displacement being very much reduced.

Lord Kelvin has said that "if a horse could gallop fast enough it could gallop over the surface of the sea without sinking in." This speed would be reached when such a number of blows were

struck by its hoofs per second upon undisturbed portions of the water surface, that the reaction due to the velocity imparted to the large amount of water put in motion was sufficient to support the horse's weight. A necessary condition of a reduction of displacement is that water should be driven downwards by the vessel. The reaction of this vertically moving water would serve to support so much of the weight of the vessel as was in excess of the displacement. The ricochetting shot rises out of the water because it imparts a downward velocity to a large quantity of water struck by it, and the equivalent of the energy required to change the direction of the shot is to be found in the downward momentum of the stricken water. But although it is possible that a lifting of the boat occurs in some degree at the speeds now attained, there seems to be little direct evidence of it.

There is doubtless a lifting above the surrounding water-level, due to the boat being upon the back of a wave which its bow frequently overhangs in such a way as to be clear of the water altogether, but it does not follow that there is a diminished displacement. It seems more probable that the improvement in performance at high speeds is due to the subsidence of the wave-making element of resistance at a critical speed, which in vessels of a similar form is proportional to the square root of the length, and which is,

therefore, correctly given by the formula already quoted.*

If the vessel is short enough in proportion to the length of the transverse system of waves produced by her passage to travel entirely or nearly so upon the back of the leading wave, the stern not extending into the following trough, she will then be moving through water having a forward velocity in the same direction as her own motion, and the frictional resistance will be reduced. This forward velocity, due to the transmission of the wave form, is, of course, much less than that in the wave of translation previously discussed, but it will be sufficient to exercise an influence upon the resistance of the vessel.

Sir Nathaniel Barnaby has well put the case in his "Watt" Anniversary Lecture in 1899, on 'Steam Speeds at Sea.' He says:—

"The propelling power in the ship is largely expended in making trains of waves. The surface water put in motion by the passage of the ship reaches the position of rest in that way, and the ship has to pay for the re-arrangement.

"The wave-making expenditure increases within certain limits at a very rapid rate as the speed of the ship increases. When the limit is reached, there is an apparent change in the behaviour of the fluid

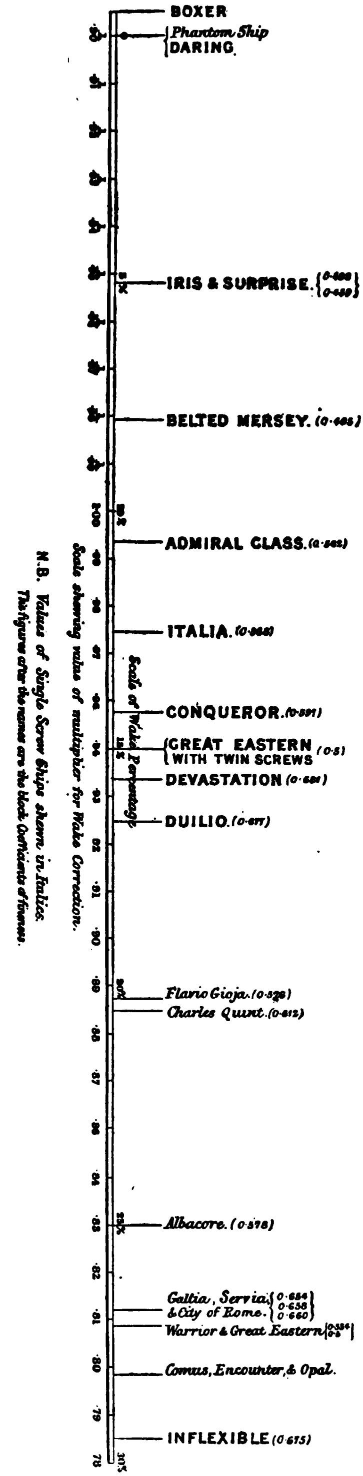
* See Sir William White's Manual of Naval Architecture, 3rd edition, 'Wave-making Resistance.'

through which the vessel is forced. In a vessel 185 feet long this change for the better begins when the vessel reaches a speed of 24 knots. The longer and heavier the ship, the higher is the speed at which nature begins to favour the engineer in his attempts to fly.

"In what way she makes this apparent change in her methods it is not easy to explain. Sir William White says, 'the boat travels upon the back slope of a wave having the same speed as herself.' She is seen to rise in the water, the bow is eventually lifted out of it, and the vessel settles down to speeds gained with comparative ease under the new conditions.

"Rails appear to be laid for the boat, length by length, before her forefoot, when the power in the control of the victorious engineer can no longer be denied."

TABLE III.



BOXER
Phantom Ship
DARING
IRIS & SURPRISE.
BELTED MERSEY.
ADMIRAL CLASS.
ITALIA.
CONQUEROR.
GREAT EASTERN WITH TWIN SCREWS (0·5)
DEVASTATION
DUILIO.
Flavio Gioja.
Charles Quint.
Albacore. (0·578)
Gallia, Servia, & City of Rome. 0·654 0·658 0·660
Warrior & Great Eastern
Comus, Encounter, & Opal.
INFLEXIBLE (0·675)
Scale of Wake Percentage
Scale showing value of multiplier for Wake Correction.
N.B. Values of Single Screw Ships shown in Italics.
The figures after the names are the block Coefficients of fineness.

67

C_B	C_A	k
150	100	18
135	109	14
123	116	10
113	125	84·
104	132	69·
97	141	55·
90	150	46·
85	158	38·
80	166	32·
76	175	28·
73	182	24·
69	191	21·
67	200	18·
64	208	16·
62	217	14·
59	225	13
57	232	12
56	240	11
		14

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CHAPTER V.

CAVITATION.

In the year 1888, speaking at the Institution of Naval Architects upon a paper by Professor Greenhill on 'A Theory of the Screw Propeller,' * the author said, "I think I may say that we who are engaged in designing and building torpedo boats and other small fast vessels, are doing pioneer work with the screw propeller, for we are among speeds which have not yet been attempted by the big ships, although they are coming close up behind us. We are, therefore, in a position to foresee possible difficulties ahead." The author proceeded to point out that one probable source of trouble in the future would arise from the fact that the velocity of the blade tips would increase as speeds increased, because the circumferential velocity of a propeller of given pitch-ratio, working at maximum efficiency, would be directly proportional to the designed speed of the ship, and he concluded by saying, "*It becomes an interesting question as to how fast the water can follow up the blades of a screw.*

* Trans. Inst. Naval Architects, xxix. pp. 343–4.

Perhaps on some future occasion we may have this point investigated. But whether there may or may not be difficulties ahead in this direction, the friction and resistance of the blades at these high velocities becomes a very serious consideration, and *it appears that it may become necessary, as the speed of vessels increases, to use screws of greater and greater pitch-ratio, and to submit to reduced efficiencies in consequence.*"

The difficulties here foreseen did not present themselves in any serious form until six years afterwards, when great trouble was experienced with the screws of H.M.S. *Daring*, a torpedo-boat destroyer, having a designed speed of 27 knots.

The difficulty, when it came, arose from the second of the two causes which the author had suggested as likely to be productive of trouble, viz. the inability of the water to flow fast enough to follow the screw-blades, and it proved to be of a formidable character.

As the screw trials of the *Daring* were very remarkable in themselves, and led to the enunciation of the theory of "Cavitation" * and to the discovery of the conditions under which cavitation would make its appearance, a chapter will be devoted to the subject.

Six pairs of screws were tried of the following dimensions, etc.

* Proc. Inst. Civil Engineers, vol. cxxii. p. 66. Thornycroft and Barnaby on 'Torpedo Boat Destroyers.'

No.	Material.	Diameter in feet.	Pitch in feet.	Surface in square feet.	Remarks.
1	Steel	6·166	8·64	8·9	Increasing pitch.
2	Bronze	6·5	9·0	8·92	„
3	„	6·166	„	„	„
4	Steel	6·375	8·3	9·0	„
5	Bronze	6·416	„	11·0	Twisted from 8 ft. uniform pitch.
6	„	6·166	8·9	12·9	Increasing pitch.

No. 3 screws were obtained by reducing the diameter of No. 2 screws. No. 4 screws were made by drawing out the blades of No. 1 screws to a larger diameter, and at the same time the blades, which were keyed into the boss, were twisted to a finer pitch. The pitch of all the screws, except No. 5, varied in the manner usual with Thornycroft screws.

It increased from the leading to the following edge, and diminished from the centre towards the tip and the root of the blade. In each case the pitch given is the mean between the forward and after edge, and the slip is calculated from this pitch. The effective pitch of the wide blade screws is somewhat greater than the mean pitch given, that is, they do not turn so fast as screws of the same diameter and mean pitch, but whose pitch is uniform instead of increasing, would do.

Fig. 35 shows the slip curves obtained with

the different screws. The performance of Nos. 1, 2 and 3 was so nearly alike that one curve only is given for them. The slip was too great at all speeds, but at 22 knots it commenced to increase very rapidly, rising to nearly 30 per cent. at 24 knots, at which speed the engines were making 384 revolutions per minute, and developing 3700 I.H.P. No. 4 screws, which had about the same surface as Nos. 1, 2 and 3, but less pitch, showed a

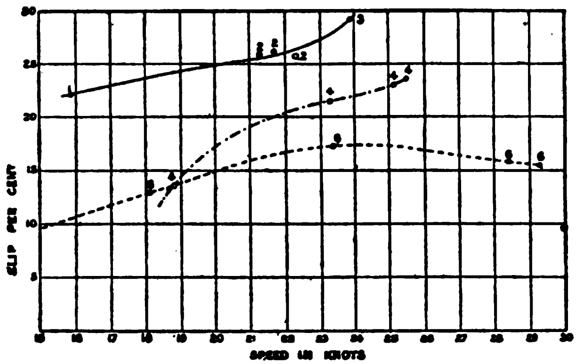

FIG. 35.—H.M.S. *Daring*—CURVES OF SLIP PER CENT. OF PROPELLERS.

marked reduction of slip at moderate speeds, but the slip curve continued to rise at speeds above 25 knots instead of falling as was expected, and at 25·5 knots, which was the highest speed that could be reached, it rose to 23·7 per cent., the engines developing 3940 I.H.P. at 407 revolutions per minute. This power, according to estimate, should have been sufficient to drive the vessel nearly a knot and a half faster, and the author felt satisfied

that the bad performance of the screws was due to the formation of cavities in the water forward of the screw, which cavities would probably be filled with air and water vapour.

This view was confirmed by the very serious vibration of the stern when the engines were driven at full speed, although when the screws were removed the engines failed to shake the vessel when run at the same number of revolutions, showing that the excessive vibration was caused by some irregular action of the propellers.

In screws Nos. 5 and 6, the surface was increased to 11 and 12·9 square feet respectively. Only one run on the knot was made with No. 5 screws, because during the second run all the blades of the port screw were lost. The estimated speed of the vessel, allowing for the tide, was 27 knots, and the slip about 18 per cent., a very considerable improvement. No. 6 screws gave a completely satisfactory performance, as shown by the slip curve marked 6, Fig. 35, and no further changes were made. They were of the same diameter as No. 3, and of practically the same mean pitch, but had 45 per cent. more surface. It will be seen that the slip is the same as that of No. 4 screws at about 19 knots, but at higher speeds is less. It reaches a maximum of 17·75 per cent. at 24 knots, and then falls to 15·75 per cent. at 29·25 knots. Comparing these screws with

No. 3, from which they differ only in blade area, it will be seen that at 24 knots the slip is reduced from 30 per cent. to 17·75 per cent. and the indicated horse-power from 3700 to 3050. The number of revolutions per minute required to obtain 24 knots with No. 3 gave 28·4 knots with No. 6. The excessive vibration disappeared.

In order to arrive at a clear understanding of what takes place, it is necessary to distinguish between the two cases—firstly, that of a propeller drawing air from the surface; and secondly, that of the formation of cavities when the propeller is submerged.

The effect upon the thrust of a fast-running screw when the blades break the surface of the water, or when air penetrates from the surface, has been already explained, see page 53, and Plate I. M. Normand has also described what he calls the rupture of the column which takes place under such conditions. He made some experiments with a moored vessel, and ascertained the thrust at different depths of immersion of the screw,* but he dealt only with the case of free communication between the atmosphere and the screw by vortices or otherwise, and showed that rupture takes place as soon as the speed at which the water is required to fill the void behind a propeller or rudder

* Very similar experiments to those of M. Normand were made by Mr. Rennie in 1878. See Proc. Inst. Naval Architects, vol. xix. p. 64.

exceeds 14·5 feet per second, or 8·5 knots at a depth of 1 metre. This is not what is called cavitation.

If the velocity which has to be imparted to the water in order that it may keep in contact with a portion of the blade situated at a depth h is less than $\sqrt{2gh}$, then, even if the blade break the surface, there will be no loss of efficiency. This is the case with the slow moving, partially submerged screws used to propel barges on canals, and it also probably explains the good results obtained with the partially submerged screws of large blade-area employed in some shallow draught tugs on the Continent. There is no reason why partially submerged screws should not give good results at any speed, provided the blade-area be sufficiently large. In the year 1862 it was proposed by Mr. Thornycroft to place twin screws in a vessel of light draft, with their shafts above the water line, so that the lower blades only propelled. The blades overlapped for nearly half the screws' diameter, and, turning in opposite directions, the side thrust of each, tending to move the stern sideways, was neutralised. Designs were prepared, but the vessels were not built. Recently a somewhat similar method of arranging screws has been employed by Mr. Barcroft, and it is understood to be working satisfactorily. The blade-area and efficiency will be governed by the same conditions as in the case of paddle wheels, to which form of propeller the

resemblance is closer than to a helical propelling surface, such as a submerged screw.

The speed at which water can follow the float of a wheel is limited by the same condition, as already described, page 7. When the screw is sufficiently submerged to prevent air from reaching it from the surface the rate at which the water can be accelerated is very much greater.

This can be illustrated as follows. Water will flow from a tank through an orifice discharging into the open air at a velocity depending upon the depth of the orifice below the surface of the water in the tank. It will flow through the same orifice into the exhausted receiver of an air-pump at a much higher velocity, depending upon the degree of exhaustion in the receiver. The velocity in the latter case will be that due to the head of water, plus the difference between the pressure of the atmosphere and that in the exhausted receiver. Similarly, the velocity with which water can be made to flow towards a submerged screw is due to the head of water over the screw, plus the atmospheric pressure, and there is consequently a definite limit to the speed to which it can attain, and a definite thrust per square inch of projected screw surface at which cavitation will commence.

A screw propels by putting water in motion sternwards. It effects its object partly by pushing the water with the after face of the blades, and partly by pulling it with the forward face.

Let it be imagined that the screw of a ship is replaced by a disc of rather less diameter than the screw, and that, instead of revolving the screw-shaft, the shaft and disc are pushed sternwards at such a speed that the disc will momentarily exert the same thrust as the screw. The propelling effect would be the same as that of the screw, provided that the movement of the disc is supposed to be very small, is confined for example to the length of the screw. So far as the action between the forward face of the screw-blades and the contiguous water is concerned, which is what it is desired to illustrate, the action of the disc affords a sufficiently close analogy. As the disc moves sternwards, it puts water in motion, not only astern of it, but also ahead of it. There being no air between the water and the forward face of the disc, a pull can be exerted upon the water, which is forced to follow the disc in the same manner that water is forced to follow the plunger of a pump.

But the pull which can be thus exerted by the disc is limited. At a little depth beneath the surface of the water, if the tension exceeds 15 lbs. per square inch, the surfaces of the disc and of the adjacent water are torn asunder, and a cavity is formed between them.

When reading a paper upon this subject before the Institution of Naval Architects in 1897,* the

* 'On the Formation of Cavities in Water by Screw Propellers at High Speeds,' by S. W. Barnaby. Trans. Inst. Naval Architects, vol. xxxix. p. 139.

author gave an illustration of cavitation, by means of a model, as follows:—

A glass cube supported upon an inclined plane of glass was supposed to represent a mass of water in advance of a screw. The force of gravity caused it to slide down the plane with a velocity depending upon the inclination given to the plane, and was an illustration of water flowing to the screw with a velocity due to the depth of the screw below the surface. The action of the anterior face of the screw-blades, or of the disc by which they may be supposed to be replaced, was represented by means of an india-rubber disc or sucker, such as is used for suspending small articles to the glass of a shop window. So long as the air was not exhausted between the sucker and the face of the glass cube, the motion of the cube down the plane could not be accelerated. It would only remain in contact with the sucker so long as this was not drawn away by the hand faster than the block could move by its own weight. This motion of the block in following the sucker represented the flow of water to the floats of a paddle wheel, and to the blades of a screw which broke the water surface.

It was shown that the action of a submerged screw could be illustrated by exhausting the air on a portion of the face of the cube by means of the sucker.

By pulling at the sucker with the hand the

cube was caused to descend the inclined plane at a greatly increased velocity due to the unbalanced pressure of the atmosphere upon its opposite face, but it was also shown that the acceleration which could be imparted to it in this way was limited. If the tension between the surfaces exceeded the atmospheric pressure they were torn apart, and thus illustrated what takes place under certain conditions with the screw, producing cavitation.

As but little more than half of the total acceleration imparted to the water by a screw is estimated to be produced by the suction of the forward surface, it might be supposed that a total thrust approaching to 30 lbs. per square inch might be obtained, but it appears that rupture occurs at parts of the screw surface long before the mean thrust per square inch of the whole surface reaches this amount.

This is probably accounted for by the fact that the thrust of portions of the screw-blade near the circumference is much greater than at portions near the boss, and also by the presence of air in sea-water which tends to expand as soon as the pressure is reduced on the forward surface of the blades.

If the slip curve of No. 6 screws be examined it will be seen that it exhibits a tendency to flatten at speeds above 27 knots, and cavitation is evidently commencing to affect the performance. The point at which the first indication of failure occurs is not marked by a sudden change but by

a flexure in the slip curve, which commences to rise rapidly after a critical speed has been passed.

The total thrust of the screw at the speed at which a change manifested itself in the character of the slip curve divided by its projected blade-area, gave a thrust of 11·25 lbs. per square inch, which is therefore about the maximum thrust which can be obtained from a screw working efficiently at a depth below the surface of 11 inches, which was the immersion of the tips of the screw blades of the *Daring*.

This figure may vary slightly with the pitch-ratio, being less if the latter is high, since the ratio which suction-thrust bears to the whole thrust varies with pitch-ratio, but the variation is so small that it may probably be neglected. It will also no doubt vary somewhat with the shape of the blade. It was obtained with elliptical blades. A higher thrust might be safe with blades which are broad at the tip.

For every additional foot of immersion the total thrust per square inch may be increased by three-eighths of a pound. Conversely, when a screw is placed in a tunnel in such a way that it rises above the level of the outside water and works in water which is supported by atmospheric pressure, the safe thrust per square inch of blade surface must be reduced by three-eighths of a pound for each foot of height of the top of the tunnel above the outside water surface.

Results in strict accordance with those of the *Daring* were obtained contemporaneously from the trials of two torpedo boats. Nos. 91 and 92 were sister vessels, and had screws of the same diameter and pitch, being made from the same drawing, but the surface of the screw of No. 92 was about 40 per cent. more than that of No. 91. At 24 knots the boat with the narrow-bladed screw required 16·5 per cent. more power than the other, and the slip of the screw of No. 91 was 20 per cent., while that of No. 92 was only 6 per cent.

The thrust in the case of No. 91 was 16·3 lbs. per square inch, and in that of No. 92 it was 11·2 lbs., the latter figure agreeing very closely with that at which cavitation commenced in the *Daring*.

By means of the very ingenious expedient of trying a model screw in water heated nearly to boiling point, the Honourable Charles Parsons has been able to reproduce and examine the phenomenon of cavitation, and has corroborated the figure first calculated by the author from the *Daring* of 11·25 lbs. as the total thrust at which it commences. More recent experiments with the *Albatross* point to the probability of this figure being if anything a little in excess of the safe thrust.

The difficulty of reproducing analogous conditions with a model lay in the high speed of revo-

lution required, but by heating the water, and thus increasing the vapour pressure, cavitation could be produced at lower speeds. Since the tension of the water vapour at boiling point is equal to atmospheric pressure, a screw would cavitate in boiling water at the same speed as it would do if air were admitted to it. In other words, steam would be given off as soon as the pressure was reduced. An analogous effect could be obtained by expanding by heat the residue of air in the exhausted receiver before referred to, page 116. Then the reduced difference of pressure between the air within and without the receiver would cause the water to flow into the latter from the tank more slowly, and, if the temperature of the air were raised sufficiently to cause a pressure inside the receiver equal to that of the atmosphere, the rate of flow would be that due to the head of water only.

The same result would be secured by using a closed tank and exhausting the air in it, so that the water would flow from one vacuous region to another, and Mr. Parsons has shown that experiments can be made in cold water, the surface being relieved from the pressure of the atmosphere.

The surface used in these calculations is the projected surface, because only the sternward component of the total thrust is considered. The improvement obtained with No. 4 screws at

moderate speeds as compared with Nos. 1, 2 and 3, see Fig. 35, which are of about the same developed area, was probably due to the greater projected area obtained by twisting the blades to a finer pitch.

More developed area is required for screws of great pitch-ratio than for those in which the pitch-ratio is small, because the projected area of a given screw is inversely proportional to the pitch-ratio.

$$\text{Projected area} = \frac{\text{developed area}}{\sqrt{1 + 0{\cdot}425\,(\text{pitch-ratio})^2}}.$$

The usual formula for the thrust of a screw is

$$\text{Thrust} \propto \mathrm{A}\, n\, p \frac{(n\, p - \mathrm{V})}{g},$$

where A = effective disc-area in square feet;
n = revolutions per second;
p = pitch in feet;
V = velocity of feed in feet per second;
g = acceleration produced by gravity = 32·2 feet per second.

This is derived from the formula for reaction or thrust already given, page 1,

$$\mathrm{R} = \frac{\mathrm{W\,S}}{g}.$$

Anp being proportional to the weight of water acted upon or W, and $np-\mathrm{V}$ being proportional to S or velocity of discharge in relation to still water.

It is not of much value for calculating the actual thrust in pounds exerted by a screw, and this for many reasons. A screw does not deliver a complete column of water at a uniform velocity, and it is difficult to estimate the proportion of the disc-area which should be considered as effective. Again, the thrust with a given diameter, pitch, and number of revolutions, is affected by the number of blades. In fact, neither the quantity nor the velocity of the water discharged can be estimated exactly from the area, pitch, and revolutions.

A fairly close approximation to the thrust is obtained by taking the developed blade-area instead of the complete disc as the value of A.

But the best way of estimating the thrust of a screw is to deduce it from the I.H.P. and the speed of the vessel.

It has already been shown (page 83) that

$$\text{E.H.P.} = \text{tow-rope resistance} \times \text{speed of ship},$$

and that

$$\text{Thrust} = \frac{\text{tow-rope resistance}}{\text{thrust-deduction factor}},^{*}$$

$$\therefore \text{Thrust} = \frac{\text{E.H.P.}}{\text{speed of ship} \times \text{thrust-deduction factor}}.$$

Using the assumptions described on page 83, viz. that the thrust-deduction and wake were each 10 per cent., so that T.H.P. = E.H.P. and hull-

* "Thrust-deduction" is the amount by which resistance is less than thrust.

efficiency* = unity, and also that E.H.P. = I.H.P. × 0·5, then

$$\text{Thrust in pounds} = \frac{\text{I.H.P.}}{\text{V}} \times 181,$$

V being speed of ship in knots; and if A = minimum projected blade area in sq. ft. which is sufficient to prevent cavitation, then

$$\text{A} = \frac{\text{I.H.P.}}{\text{V}} \times 0{\cdot}112.$$

The general form of the equation would be

$$\text{A} = \frac{\text{E.H.P.}}{\text{V} \times \text{thrust-deduction} \times \text{thrust in pounds per sq. in.}} \times 2{\cdot}26.$$

Cavitation will probably be the cause of considerable difficulty in the future. Although no general solution of the problem can be said to have been found in the plan adopted in the *Turbinia*, of dividing the power among a number of propellers, several of which are upon the same shaft, it is quite possible that this expedient will serve for a time to permit of higher speeds being obtained than would otherwise be practicable, and at any rate multiple propellers would seem to be the probable direction in which relief will be sought, since the reduction in diameter of screw admits of deeper immersion of the blade-tips, and

* "Hull efficiency" $= \frac{\text{E.H.P.}}{\text{T.H.P.}} = \frac{\text{tow-rope resistance} \times \text{V}}{\text{thrust} \times \text{V} \times \text{wake-factor}}$ $= \frac{\text{thrust-deduction factor}}{\text{wake factor}}$. In nearly all cases thrust-deduction factor = wake factor, and "hull efficiency" = unity.

therefore of a higher thrust per unit of surface, and it admits also of higher rates of revolution and a saving in the weight of machinery.

The *Turbinia* was first constructed with one turbine and one propeller, and was well advanced towards completion before the publication of the results of the trials of the *Daring*, which first drew attention to the effect of cavitation, and although these showed that in all probability trouble would be experienced, it was not anticipated that it would be so serious as it proved. The special difficulty in the *Turbinia* arose from the extremely high rate of revolution at which the Turbo-motor had to run, which made it necessary to employ a screw of very small relative diameter, and therefore of small blade-area. With extreme boldness and ingenuity Mr. Parsons devised the plan of sub-dividing the power among nine screws disposed on three shafts, and he met with complete success, but, as has already been said, although tending to mitigate, it does not remove the main difficulty to be faced, which is the undue growth of blade area with augmentation of speed. See note at end of chapter.

It has been stated, see page 109, that if a given pitch-ratio be selected, and a screw designed to work at the slip which gives maximum efficiency, i.e. about nine abscissa value, the circumferential velocity of the blade-tips is proportional to the speed of the ship.

It is easy to show, similarly, that under the same conditions it does not matter whether the ship is a big ship or a little ship, whether the horse-power is large or small, or whether the ship is driven with one large propeller or half a dozen little propellers, the thrust per square inch of surface will be proportional to the *square of the speed* of the ship. This can be shown by taking out, by means of the constants in Table II., the disc-area of a screw for a given power and speed, and also of a similar screw for the same power and twice the speed, and calculating the thrust per square inch of surface in each case. It will be found to be four times as great in the case of the screw which has a velocity of feed twice as great as the other. See examples, page 132.

It has been shown that with given pitch-ratio and efficiency, the disc-area per unit of H.P. is inversely proportional to the cube of the speed, see page 81, and that the total thrust of the screw with a given E.H.P. is inversely proportional to the speed, see formula for thrust, page 124. It follows that for twice the speed there would be but one-eighth the total propelling surface and one-half the total thrust, and the thrust per unit of surface must therefore be four times as great.

The two equations referred to above for disc-area and thrust, afford means for calculating the speed at which the thrust of any given propeller, of the type used in the model experiments and

situated at the same depth below the water surface as the screws of the *Daring*, will begin to be affected by cavitation.

Let A = disc-area in square feet;
P = indicated horse-power per screw;
V = speed of vessel in knots at which cavitation commences;
C_A = constant for disc-area from Table II.

Then (1)

$$A = C_A \times \frac{P \times \text{propulsive coefficient correction} \times \text{blade correction}}{V^3 \times (\text{wake correction})^3}.$$

Assume a ten per cent. thrust-deduction, and a propulsive coefficient of $0 \cdot 5$. Assume also that the projected blade area equals $A \times 0 \cdot 207$, which would be about the ratio in the case of a three-bladed screw of $1 \cdot 2$ pitch-ratio having elliptical blades of width $= \frac{\cdot 4\, D}{2}$.

Other values can be substituted for any of these assumptions to suit different cases (see examples on page 133).

Taking the thrust per square inch at speed V as $11\frac{1}{4}$ lbs., and the total thrust as equal to $\frac{\text{effective horse-power}}{\text{speed of ship} \times \text{thrust-deduction}}$, we get

$$(2) \qquad V = \frac{0 \cdot 54\, P}{A}.$$

Substituting the value of A from (1)

$$V = \frac{\sqrt{C^A}}{\cdot 683}.$$

The general form of the equation would be

$$V = \sqrt{\frac{C_A \times \text{thrust-deduction} \times A_P \times T_0 \times \text{blade correction}}{1\cdot125 \times (\text{wake correction})^3}},$$

where T_0 = thrust in pounds per sq. in. of projected area;

A_P = the value of the ratio $\frac{\text{projected area}}{\text{disc area}}$.

For blade correction, see page 86.

For wake correction, see Table III.

The values of V calculated from this formula for screws of different pitch-ratios and slip-ratios are given in Table IV., page 138.*

Cavitation can only be avoided as speeds increase, by immersing the screws more deeply or by enlarging the blade-area above that which would be considered suitable if cavitation could be neglected. It is not usually convenient to increase the immersion, since it involves either an increase in the draught of water of the ship, or the employment of multiple screws.

When a blade of "standard" width gives insufficient surface to prevent cavitation, there are three ways by which the surface may be augmented.

1. The blades may be made wider, other things remaining the same.

2. A lower abscissa value may be used, the constants being taken from columns 8, 7, 6 or 5, in Table II.

* The projected area of the standard screw at different pitch-ratios has been calculated from the formula on page 123.

This entails the use of a larger diameter of propeller working at less slip than corresponds to maximum efficiency.

3. A larger pitch-ratio may be chosen, which will carry with it an increase in diameter and a reduction in the rate of revolution, and therefore a probable augmentation to the weight of machinery.

It will be seen, by looking at Table IV., that increasing the pitch-ratio is less efficacious than reducing abscissa value. This is due to the fact that increasing the pitch-ratio reduces the ratio of projected-area to disc-area.

Either of the three courses tends to a waste of power if carried to an extreme.

It is not to be supposed that cavitation imposes a limit to the speed which can be obtained by means of screws. Much higher speeds than any yet reached will doubtless be achieved, but the efficiency of propulsion will suffer diminution after certain limits have been exceeded.

It will be seen, by reference to Table IV., that a three-bladed screw of standard blade width, designed to run at the slip corresponding to maximum efficiency, and with normal wake and thrust deduction, commences to feel the effects of cavitation at a speed varying from 23·0 knots to 27·4 knots, depending upon the pitch-ratio. As the screws of torpedo boats have usually been made with a much greater proportion of blade-area to

disc-area than the "standard" screw, they have been able to run at higher speeds without trouble.

The speeds in columns 16 and 17 appear to be very low, but it must be remembered that screws working at such a high slip-ratio as this would be very unusual, and the diameter being so very small for the work to be done, no one would think of making the blades so narrow as $\frac{\cdot 4\,D}{2}$, because experience has shown that a certain limiting area must be provided.

If the width of blade in proportion to diameter is increased until the projected surface is equal to four-tenths of the disc-area, a ratio which can be obtained even with three-bladed screws of moderate pitch-ratio, the cavitating speed can probably be raised to above 40 knots without greatly affecting the efficiency, but multiple screws would almost certainly be required, in order to admit of the employment of sufficiently fast-running machinery.

The limiting width of three-bladed screws is reached at from 32 to 33 knots, with the pitch-ratio and wake-value usual in Destroyers. With four-bladed screws the cavitating speed can be still further raised.

To the after end of the propeller boss a tapering cone of considerable length, say from 1 to $1\frac{1}{2}$ times the length of the boss, should be fitted in

very high speed vessels to prevent the formation of a cavity in the water behind it.

Examples from Table II., proving that the Thrust per Unit of Surface varies as the Square of the Speed.

1. Find the disc-area of a screw of $1 \cdot 2$ pitch-ratio to work at maximum efficiency. Speed of vessel 20 knots. I.H.P. 6000. Ratio of projected blade-area to disc-area to be as $0 \cdot 3$ to 1.

The disc-area constant for these conditions is 288; and

$$\text{Disc-area} = 288 \times \frac{6000}{20^3} = 216 \text{ square feet.}$$

Projected blade area $= 216 \times \cdot 3 = 64 \cdot 8$ square feet:

Thrust in pounds per square inch of projected surface

$$= \frac{0 \cdot 5 \times 6000 \times 33{,}000}{20 \times 101 \cdot 33 \times 0 \cdot 9 \times 64 \cdot 8 \times 144} = 5 \cdot 8.$$

2. Find the disc-area of a screw of the same model and same abscissa value, for a vessel of the same power but of 40 knots speed.

$$\text{Disc-area} = 288 \times \frac{6000}{40^3} = 27 \text{ square feet.}$$

Projected blade-area $= 27 \times \cdot 3 = 8 \cdot 1$ square feet.

Thrust in pounds per square inch of projected surface

$$= \frac{0 \cdot 5 \times 6000 \times 33{,}000}{40 \times 101 \cdot 33 \times 0 \cdot 9 \times 8 \cdot 1 \times 144} = 23 \cdot 2,$$

or four times the thrust per inch of the screw for 20 knots.

To show that the thrust per unit of surface is the same whether the ship be driven by one screw or by two screws.

3. Find the disc-area of twin-screws for the 20-knot ship of 6000 I.H.P. in the preceding examples.

$$\text{Disc-area} = 288 \times \frac{3000}{20^3} = 108 \text{ square feet.}$$

$$\text{Projected blade-area} = 108 \times \cdot 3 = 32 \cdot 4 \text{ square feet.}$$

Thrust in pounds per square inch of projected surface,

$$= \frac{0 \cdot 5 \times 3000 \times 33{,}000}{20 \times 101 \cdot 33 \times 0 \cdot 9 \times 32 \cdot 4 \times 144} = 5 \cdot 8,$$

which is the same thrust per inch as in (1), when a single screw was used.

Examples in the Use of the Formula for Cavitating Speed on page 129.

1. Find the cavitating speed of a three-bladed screw of "standard" width, 1·2 pitch-ratio, 9 abscissa value, 6 per cent. thrust deduction, 6 per cent. wake, 1·15 blade correction, and 1 foot immersion of blade tips.

$$V = \sqrt{\frac{288 \times \cdot 94 \times \cdot 207 \times 11 \cdot 25 \times 1 \cdot 15}{1 \cdot 125 \times (1 \cdot 04)^3}} = 23 \cdot 9 \text{ knots.}$$

2. Find the cavitating speed of a four-bladed screw of "standard" width, 1·8 pitch-ratio, 8 abscissa value, 8 per cent. thrust deduction, 8 per cent. wake, and 5 feet immersion of blade tips

The permissible thrust per sq. in. of projected blade area will be

$$11 \cdot 25 + (4 \times \tfrac{3}{8}) = 12 \cdot 75 \text{ pounds.}$$

$$V = \sqrt{\frac{468 \times \cdot 92 \times \cdot 226 \times 12 \cdot 75}{1 \cdot 125 \times (1 \cdot 02)^3}} = 32 \cdot 3 \text{ knots.}$$

NOTE.

In order to show the special difficulty which has to be faced in the design of a screw propelled by a steam-turbine, it may be interesting to work out an example.

Let a screw be required for a speed of 31 knots and 1575 I.H.P.

Suppose the propulsive coefficient to be 0·6 and the wake and thrust deduction 6 per cent.

Let it be supposed at first that the number of revolutions is not fixed.

At 1·2 pitch-ratio and 9 abscissa value we find from Table II.

$$C_A = 288 \text{ and } C_B = 92,$$

which will give a disc-area of 16·2 square feet, a diameter of 4·47 feet, and 662 revolutions.

A projected area of 6·5 square feet will be found requisite, in order to limit the thrust per inch to 11·25 lbs. The projected area will consequently amount to 40 per cent. of the disc-area. The blades would be wide, but otherwise the screw would be a normal one.

But if the revolutions are fixed at about 2000 per minute, to suit a steam-turbine, very different proportions are required. If it be desired to put all the power through a single screw it will be found necessary to take the constants from Table II.,

corresponding to a pitch-ratio as fine as 0·8, and an abscissa value nearly as high as 15, in order to get a screw which will run fast enough. The disc-area would be only 5 square feet, and since the minimum projected area to avoid cavitation is 6·5 square feet, or considerably greater than the disc-area, it is obvious that it would be impossible to avoid a very excessive thrust per unit of surface, and it is easy to understand the enormous effect cavitation had upon such a vessel as the *Turbinia*, as at first constructed.

Now suppose the total power to be divided among nine propellers giving 175 I.H.P. per screw.

It will be found that a pitch-ratio of 1·18 and an abscissa value of 9 will give a propeller of 1·78 square feet disc-area and 1·5 feet diameter, which will run at 2000 revolutions. The projected area for a thrust of 11·25 lbs. per square inch will be 0·72 square feet, or exactly one-ninth of that required for the single screw, and the ratio of projected area to disc-area will be the same as for the single screw, and the blades will require to be of the same proportionate width.

When screws are placed one behind another upon the same shaft it does not seem probable that all can be equally efficient. Unless the distance between them is very great it is to be expected that the aftermost screws will be affected by the race of those ahead of them. There is little doubt

that the pitch of the middle screw when three are placed on the same shaft should be greater than that of the foremost screw, and the pitch of the aftermost, greater than either, but the difficulty of estimating the velocity of feed of each is very great, and in the *Turbinia* the screws were all made alike.

TABLE IV.—SHOWING SPEED AT WHICH CAVITATION

Pitch-ratio.	Efficiency 63 per cent.	Efficiency 66 per cent.	Efficiency 67 per cent.	Efficiency 68 per cent.	Efficiency 69 per cent.	Efficiency 69 per cent.	Efficiency 69 per cent.
	knots	knots	knots	knots	knots	knots	knots
0·8	33·9	30·0	27·3	24·9	23·0	21·1	19·6
0·9	34·7	30·7	27·9	25·5	23·5	21·6	20·0
1·0	35·4	31·4	28·4	26·1	24·0	22·1	20·4
1·1	36·0	32·0	29·0	26·6	24·5	22·5	20·8
1·2	36·5	32·5	29·5	27·0	24·9	22·9	21·2
1·3	37·0	33·0	29·9	27·4	25·3	23·3	21·5
1·4	37·5	33·4	30·3	27·7	25·6	23·6	21·8
1·5	37·9	33·8	30·7	28·0	25·9	23·9	22·1
1 6	38·3	34·2	31·0	28·3	26·2	24·1	22·3
1·7	38·6	34·5	31·3	28·6	26·4	24·3	22·5
1·8	38·9	34·7	31·5	28·8	26·6	24·5	22·7
1·9	39·1	34·9	31·7	29·0	26·8	24·6	22·8
2·0	39·3	35·1	31·9	29·1	26·9	24·7	22·9
2·1	39·5	35·3	32·0	29·2	27·0	24·8	23·0
2·2	..	35·5	32·1	29·3	27·1	24·9	23·1
2·3	..	35·6	32·2	29·4	27·2	25·0	23·2
2·4	..	35·7	32·3	29·5	27·3	25·1	23·3
2·5	..	35·8	32·4	29·6	27·4	25·2	23·4
	5	6	7	8	9	10	11

ABSCISSA

The constant V_0 is the formula

COMMENCES WITH THREE-BLADED SCREWS OF "STANDARD" AREA.

Efficiency 69 per cent.	Efficiency 68 per cent.	Efficiency 67 per cent.	Efficiency 66 per cent.	Efficiency 64 per cent.	Efficiency 63 per cent.	A_P	V_C
knots	knots	knots	knots	knots	knots		
18·2	16·7	15·7	14·5	13·5	12·6	·232	$\frac{\sqrt{C_A}}{\cdot 639}$
18·6	17·1	16·0	14·8	13·8	12·9	·226	$\frac{\sqrt{C_A}}{\cdot 650}$
19·0	17·5	16·3	15·1	14·1	13·2	·220	$\frac{\sqrt{C_A}}{\cdot 661}$
19·3	17·8	16·6	15·4	14·4	13·5	·213	$\frac{\sqrt{C_A}}{\cdot 672}$
19·6	18·1	16·9	15·7	14·6	13·7	·207	$\frac{\sqrt{C_A}}{\cdot 683}$
19·9	18·4	17·1	15·9	14·8	13·9	·200	$\frac{\sqrt{C_A}}{\cdot 694}$
20·1	18·6	17·3	16·1	15·0	14·1	·194	$\frac{\sqrt{C_A}}{\cdot 705}$
20·3	18·8	17·5	16·3	15·2	14·2	·188	$\frac{\sqrt{C_A}}{\cdot 717}$
20·5	19·0	17·7	16·5	15·3	14·3	·181	$\frac{\sqrt{C_A}}{\cdot 729}$
20·7	19·2	17·8	16·6	15·4	14·4	·176	$\frac{\sqrt{C_A}}{\cdot 741}$
20·9	19·3	17·9	16·7	15·5	14·5	·170	$\frac{\sqrt{C_A}}{\cdot 753}$
21·0	19·4	18·0	16·8	15·6	14·6	·165	$\frac{\sqrt{C_A}}{\cdot 765}$
21·1	19·5	18·1	16·9	15·7	14·7	·159	$\frac{\sqrt{C_A}}{\cdot 777}$
21·2	19·6	18·2	17·0	15·8	14·8	·155	$\frac{\sqrt{C_A}}{\cdot 789}$
21·3	19·7	18·3	17·1	15·9	14·9	·150	$\frac{\sqrt{C_A}}{\cdot 801}$
21·4	19·8	18·4	17·2	16·0	15·0	·145	$\frac{\sqrt{C_A}}{\cdot 813}$
21·5	19·9	18·5	17·3	16·1	15·0	·142	$\frac{\sqrt{C_A}}{\cdot 825}$
21·6	20·0	18·6	17·3	16·1	15·0	·137	$\frac{\sqrt{C_A}}{\cdot 837}$
12	13	14	15	16	17		

VALUES.
for the cavitating speed.

CHAPTER VI.

GEOMETRY OF THE SCREW.

IF a point move on the surface of a cylinder in such a way that, while moving uniformly around the cylinder it advances uniformly in the direction of its axis, it will trace a curve known as the helix. Imagine the cylinder cut on one side by

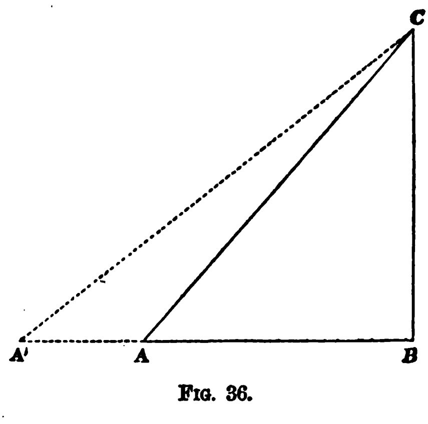

FIG. 36.

a straight line parallel to the axis meeting the helix in consecutive points A C, Fig. 36, and then unrolled and laid flat, the circumference through A will become the straight line A B; B C at right angles to it will represent the direction of the axis, while that part of the helix formed during

a complete revolution of the tracing point will be represented by the line A C. Since the distances moved by the point in the directions A B, B C are proportional, A C is a straight line. B C being the distance moved through in the direction of the axis, while the point goes once entirely round the cylinder, is the pitch; while the angle B A C, which the unrolled helix makes with a plane at right angles to the axis, is the angle of the helix or screw.

If a straight line move uniformly round an axis which it intersects, and to which it is always at right angles, advancing at the same time uniformly in the direction of the axis, it will sweep out a surface known as a helicoid, and every point in the generating line will trace a helix as described above, necessarily lying on this helicoid.

Since during a complete revolution of the generating line every point moves through the same distance in the direction of the axis, the helicoid is a surface of uniform pitch, that is, B C, Fig. 36, is constant for the helices traced by all points in the generating line.

A helicoid can therefore be, and often is, used for the acting surface of a screw-blade of uniform pitch.

It is not necessary, however, that the generating line should be at right angles to the axis; such a surface may be generated by any line, straight or curved, moving uniformly along

and revolving uniformly around an axis, intersecting and always making the same angle with it.

The helix traced by any point in the generating line will also be the curve of intersection with the screw surface of a co-axial cylinder of radius equal to the perpendicular distance of the point from the axis. The larger the radius of the cylinder, the larger of course the length of the circumference, as A′B, Fig. 36, and as the pitch is constant, it follows that the angle of the helix must decrease as the radius of the intersecting cylinder increases. We thus arrive at the fundamental geometrical property of a surface of uniform pitch, viz.: Co-axial cylinders intersect it in helices, all of which have the same pitch, but whose angles vary, decreasing as the radius of the cylinder increases. Near the axis, therefore, the helices will approximate in direction to that of the axis, and as the distance from the axis increases, they will lie more and more at right angles to it. If θ be the angle of the helix, p the pitch, and r the radius of the intersecting cylinder,

$$\tan \theta = \frac{p}{2 \pi r}.$$

The curve of intersection of a co-axial cylinder with a screw surface will hereafter be referred to briefly as the "curve of intersection," or the "helix of intersection."

In commencing the design of a propeller for a given ship, the diameter and pitch must be first

decided, and the considerations which will determine these having been fully dealt with in the preceding chapter, it is only necessary to say that it is usual to provide for an immersion of the tip of the upper blade equal to about one-tenth the diameter of the propeller, and to allow a clearance from the ship's side varying from six inches in small ships to one foot in large ones.

With regard to the number of blades to be used, if the necessary disc-area can be obtained with a three-bladed propeller it is to be preferred, but if the draught of water is limited, a four-bladed propeller is practically equivalent to a three-bladed one of rather larger diameter.

The expanded blade-area of the model screws described in the preceding chapter, and which have been called "standard" screws, is an ellipse having a major axis equal to the radius of the propeller, and a minor axis equal to four-tenths the major axis. It is often found, however, that on account of the diameter being limited, sufficient blade-area cannot be obtained with these proportions; in such cases the elliptical form is adhered to, with an increased minor axis of from ·5 to ·55 the major axis.

If sufficient area cannot thus be obtained even with four blades, as in the case of shallow draught vessels when the diameter is very limited, the elliptical form may be departed from, and the blade may be widened at the tip. A boss of dia-

meter equal to $\frac{\text{diameter of screw}}{3\frac{1}{4} \text{ to } 3\frac{1}{2}}$ will cut away about one-fifth the area of the standard ellipse. A boss of this size is adapted to the Griffiths form of screw with adjustable blades. When a propeller is cast solid, a boss equal to one-fifth the diameter of the screw is sufficient, and this will cut away about one-eighth of the area of the standard ellipse.

The expanded blade-area, which may be described as a flat surface of approximately equivalent area to that of the blades, both as to amount and disposition, is derived as follows:—

A co-axial cylinder will intersect the screw surface in a helical curve, making a certain angle with the axis, and it will intersect a plane passing through that diameter of the cylinder which passes through the middle point of the helical curve, and making the same angle with the axis in an elliptical arc.

The length of the screw being small compared with the pitch, these two arcs will nearly coincide, and no great error will be involved by assuming that they do coincide. Imagine these elliptical arcs at all radii to be swung round a common centre line till they all lie in the same plane, with their major and minor axes respectively coincident (though necessarily of different length), then the curve passing through their extremities will form the boundary of the expanded blade-area. This

area is very nearly equal to the actual whole surface of the acting face of the blade, being, in fact, something less than it.

To draw a right-handed uniform-pitch screw of Admiralty or Griffiths type with elliptical blades of standard form, with generating line at right angles to the axis, with axis horizontal and centre line of blade upright:—

Draw a straight line A B, Fig. 37, Plate III., to represent the axis of the propeller, and B C at right-angles to it equal to the radius of the propeller. With B C as major axis, and minor axis equal to $\frac{4}{10}$ of B C, describe the ellipse B D C E. Draw the circle F H G of radius equal to that of the boss, cutting the ellipse in F G; the area G D C E F H G is the expanded area.

Divide H C into a number of parts, preferably equal, at the points *a*, *b*, *c*, *d*, *e*, *f*. If p be the constant pitch of the propeller, take B A equal to $\frac{p}{2\pi}$, and join A H, A *a*, A *b*, A *c*, etc.; these lines are called the pitch-lines.

Then since the tangent of the angle which any one of these lines, passing through a point on B C distant r from B, makes with B C is $\frac{p}{2\pi r}$, these angles are the angles of the corresponding helices of intersection, and, therefore, by the assumption previously explained, they are the angles which the planes of the elliptical arcs forming the ex-

panded area make with the plane at right angles to the axis of the propeller when the arcs approximately coincide with the corresponding helices on the actual blade. Consider that elliptical arc on the expanded area passing through *c*, B*c* is its semi-minor axis, and since the angle *c* A B is its inclination when lying in its position on the actual blade to the axis of the intersecting cylinder, *c* A must be the length of its semi-major axis. Similarly A H, A *a*, A *b*, etc., are the lengths of the semi-major axes of the elliptical arcs through H, *a*, *b*, etc., respectively, and B H, B *a*, etc., are the corresponding semi-minor axes. It follows at once that A is a focus of all these elliptical arcs, the other focus being at K, where B K = B A.

Draw, therefore, through the points H, *a*, *b*, etc., elliptical arcs with foci A, K, and semi-major axes respectively equal to A H, A *a*, A *b*, etc. Then by the assumption, these elliptical arcs represent the helical arcs of intersection turned about B C till they lie in the same plane. If we reverse the process, and turn them back from the expanded area through the same angle, their extremities will give us points on the outline of the blade.

The angle any particular arc must be turned through is that which its pitch-line makes with B A.

This enables us to draw the projections, and in dealing with these we shall take a particular one of the elliptical arcs, and show how to obtain the

projections of the point at one of its extremities; the projections of the other extremity may be obtained in a similar way, but on the other side of the centre-line. This will be a specimen arc.

If the same method be adopted for the extremities of all the other arcs, series of points will be obtained, and if fair curves be drawn through the respective series, we have the required projections.

For each projection of the blade we shall, therefore, deal only with the projection of one point on its outline.

Take any one of the elliptical arcs as the specimen, say that through c: viz. $c_1\,c\,c_2$, Fig. 37, Plate III. Draw $c_2\,c_0$, parallel to the axis, mark off on the pitch-line through c, $c\,c_3 = c_0\,c_2$. Draw $c_3\,c_4$ perpendicular to A B and $c\,c_4$ parallel to it. Then $c\,c_4$ is the projection of $c_0\,c_2$ on a fore and aft plane, and $c_3\,c_4$ is its projection on an athwartship plane. Set off $c_0\,c_5 = c_3\,c_4$, then c_5 is a point on the athwartship projection of the blade. Similarly on the other side of the arc we get another point c_6, and so on for all the other arcs.

It may be noticed that this projection may also be found thus. Draw circular arcs with B as centre through H, a, b, c, etc. On the blade the elliptical arc approximately coinciding with the helical arc will project on an athwarthship plane into a circular arc, the chord of the former, of

course, projecting in that of the latter. Dealing with the arc through c, draw $c_0 c_2$ parallel to the axis through c_2, meeting the circular arc in c_5, then the elliptical arc $c\,c_2$ will project on an athwartship plane into the circular arc $c\,c_5$. c_5 is, therefore, a point on the athwartship projection of the blade.

For the fore and aft projection take K P, Fig. 38, as the axis of the propeller, and draw P Q at right angles to it. Set off distances P H_7, P a_7, P b_7, etc., respectively equal to B H_0, B a_0, B b_0, etc., and draw straight lines through the points H_7, a_7, b_7, etc., parallel to K P. Dealing with the line $c_9\,c_7\,c_8$, set off $c_7\,c_8$ equal to $c\,c_4$, Fig. 37, then c_8 is a point on the fore and aft projection of the blade. Similarly for other points.

For the horizontal projection take N R, Fig. 39, as the axis, R being forward, and draw straight lines through N, making angles with N R equal to H A B, a A B, etc., Fig. 37, and inclining as shown, since the blade is right handed. These straight lines represent the directions on the actual blade of the chords of the elliptical arcs, the lengths are given in Fig. 37. Dealing with the line N c', which is the direction of the chord $c_0\,c_2$, Fig. 37, mark off N $\gamma = c_0\,c_2$, Fig. 37, then γ is a point on the horizontal projection. On the other side of N we get similarly γ', the horizontal projection of the other extremity of the chord. Similarly for other points.

The complete projections of a three-bladed propeller are shown in Figs, 42, 43, 44, Plate IV.

The athwartship projections of the two lower blades are simply repetitions of that of the upper blade, their centre lines being inclined to that of the upper one at angles of 120°, Fig. 42.

The fore and aft projection of the lower blade, Fig. 43, is obtained thus. Across the projections of the blades in Fig. 42 and the top blade of Fig. 43, draw straight lines at right angles to their central lines, as in Figs. 37 and 38. Consider the point whose athwartship projection is c_2 on the right-hand lower blade, Fig. 42; this being on the following edge will be abaft the centre line in Fig. 43, the corresponding point on the leading edge (c_1) appearing on the forward side.

Then to obtain the position of this point on the fore and aft projection, it must be borne in mind that it must lie in the same vertical plane, perpendicular to the axis, as it does when the blade is upright, and at the same distance from the horizontal plane through the axis as in its athwartship projection.

Consider the point whose athwartship projection is c_2, Fig. 42. Draw $c_3\,c_4$, Fig. 43, on the left-hand side of the centre line P Q, parallel to the axis, at the same distance from it as c_2, Fig. 42, is from B P. Take $c_3\,c_4$ equal to $c_0\,c_7$ (c_7 being the position on the fore and aft projec-

tion which the point would occupy if the blade were upright), then c_4 is a point on the fore and aft projection of the right-hand lower blade. Similarly for other points.

Proceed in a similar manner for the other lower blade, bearing in mind that the upper edge, Fig. 42, is here the leading or forward edge. The projection of one blade only is shown in Fig. 43 to avoid confusion.

In the horizontal projection of the lower blades, Fig. 44, we proceed similarly. The point whose athwartship projection is c_2, Fig. 42, for example, will appear on the after side of the centre line of the blade at a distance from it equal to $c_0\ c_7$, Fig. 43; and from the axis equal to the perpendicular distance of c_2, Fig. 42, from B E. The projections of a three-bladed propeller are thus completely determined. It may be remarked that it is often considered sufficient to replace the elliptical arcs on the expanded area, Fig. 37, by circular arcs with B as centre, forming the projections from the chords of these arcs precisely as has been described for the elliptical ones.

The error introduced by this method of procedure is not great, being only appreciable towards the root of the blade, where it is of little consequence.

Blades are sometimes made with the generating line inclined to the axis, or in technical terms, they are made with a skew.

Let the two straight lines A B, A C, Fig. 45, the former at right angles to an axis, the latter inclined to A B at an angle a, moving together, generate screw surfaces of equal form and equal pitch. Then the helices of intersection of these two surfaces will be exactly similar, and one will be always a constant distance from the other, this distance being at a radius r, $r \tan a$. Imagine these two surfaces so far similar, that when A C at any radius leaves the surface, A B at the same radius leaves its surface, then the expanded area of the surface so formed by A B will represent what may be termed the effective expanded area of that formed by A C, and this should be of the elliptical or other form which would have been used if the generating line had been at right angles to the axis.

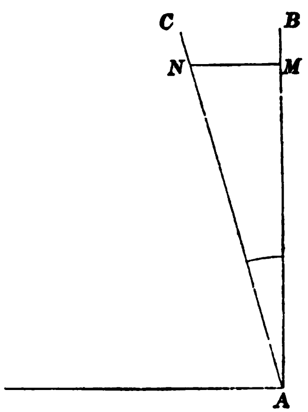

FIG. 45.

A blade generated by A C would therefore be formed from a blade generated by A B, simply by setting the helices of intersection definite distances aft, the distance at M for example being M N. It follows, therefore, that the athwartship projection of the blade for the same "effective" expanded area is independent of the skew, and consequently

for a skew blade with effective expanded area, as for the blades shown in Figs. 42, 43, 44, the athwartship projection will be as in Fig. 42, no matter what the skew be.

The fore and aft projection of the upper blade, Fig. 46, will be formed by using a centre line inclined to the vertical at an angle equal to the inclination of the generating line to the vertical, and proceeding as in Fig. 38, setting off the distances horizontally. The projections of the lower blades are determined in a similar way to that described for Fig. 43. For example, the point whose athwartship projection is c_2, Fig. 42, will appear on Fig. 46 at c_4, lying in the same vertical straight line as c_7 (c_7 being the position of this point when the bade is upright), and being perpendicularly away from the axis a distance equal to that of c_2 from the horizontal plane through the axis.

Next consider the horizontal projections, Fig. 47.

For the top blade the pitch-lines are not all drawn through the same point as in Fig. 44, but each line is drawn at the corresponding angle to the axis through a point on the axis at a distance from $M_1 N_1$ (corresponding to M N in Fig. 46) equal to the distance of the corresponding point on the generating line P Q from M N, Fig. 46, the process then being as in Fig. 39. For the lower blade we proceed as in Fig. 44; for instance, the

point c_2, Fig. 42, corresponding to c_7, Fig. 46, when the blade is upright, will appear in Fig. 47 at c_8 at a distance from the axis equal to the distance of c_2, Fig. 42, from the vertical plane through the axis, and from $M_1 N_1$ equal to the distance of c_7 from $M N$, Fig. 46.

The projections of a three-bladed propeller with skew blades are shown in Figs. 42, 46, 47, except that the left-hand lower blade of Fig. 42 has not been shown in Fig. 46 to avoid confusion.

When the generating line is curved, the method is now obvious.

We pass on to consider blades not of uniform pitch.

If it is desired to make the pitch increase from the root of the blade towards the tip, that is, if the pitch at c, Fig. 37, for example, be $2\pi B Z$ instead of $2\pi B A$, then the pitch-line at c will be $Z c$ instead of $A c$. With this modification the method of constructing the projections is the same.

Blades are frequently made with an increasing pitch from leading to following edge. In Fig. 48 let the pitch at the leading edge, at a radius equal to $\frac{c b}{2\pi}$, be $a b$, and suppose as we go to a point d the pitch increases to $b e$, then $e c b$ will be the pitch-angle at this point. Similarly, if at f the pitch increases to $b g$, then $b c g$ is the pitch-angle at f. Draw $d h$ parallel to $c e$, and $f k$ parallel to $c g$, then if the curve of intersection at the radius

under consideration be unrolled, as in Fig. 36, instead of being straight, it will be as *a d f*, in other words, the pitch-line through *a* will not be straight; if the pitch varies continually, then the pitch-line will be a continuous curve.

A blade of uniformly varying pitch from leading to following edge would be generated by a line always intersecting, and always inclined at the same angle to, the axis, moving uniformly

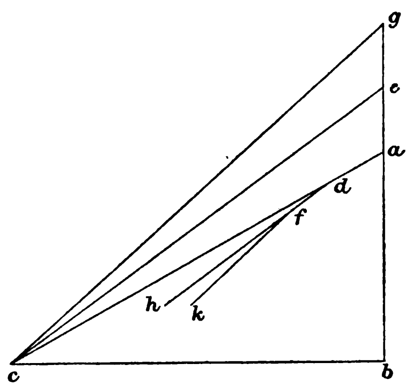

FIG. 48.

round the axis while advancing with uniform acceleration along it.

Only the acting face of the blade preserves the helical form, the back being made to give the required thickness at the different parts. The thickness at the centre line of the blade is first fixed at root and tip, and set off from the face of the blade. The two points thus found being joined, the distance of this line from the face gives the thickness at any

intermediate point (see Fig. 38). The thickness at the tip should be as small as possible, consistent with good casting or forging as the case may be, and in gun-metal is generally made about $\frac{3}{64}$ inch per foot of diameter. The blade, when acting on the water, is in the position of a beam under a load distributed all over its surface, varying in intensity, but it would be very difficult to find the bending moment at the root, and it is usual to make the rough assumptions that the total pressure on the blade tending to break it about the root section is proportional to the indicated thrust or to $\frac{P}{p\,R}$, where

$$\begin{aligned} P &= \text{I.H.P. per blade}; \\ p &= \text{the pitch of the propeller}; \\ R &= \text{the revolutions per minute}; \end{aligned}$$

also that the "leverage" is proportional to $D - d$, D being the diameter of the propeller, and d that of the boss; and that the moment of inertia of the root section of breadth b and depth h is proportional to $b\,h^3$, then h, the required thickness at the middle of the root, is obtained from the formula

$$h^2 = \frac{c \,.\, P\,(D - d)}{p \,.\, R \,.\, b};$$

the value of the coefficient c being about 230 for gun-metal, and 90 to 100 for forged steel or manganese bronze blades. In this formula p, D

and d are to be taken in feet, and b and h in inches.

Although this formula is satisfactory for screws of moderate pitch-ratio and slip-ratio, it does not take account of all the conditions, and is less suited for general application than the following, which has been devised by Mr. A. E. Wild, and used for some time in the design of screws for which the author has been responsible. The following are the assumptions made :—

Let it be assumed that T.H.P. = E.H.P. = ·58 I.H.P.
Then the fore and aft thrust per blade

$$= \frac{\cdot 58\,P \times 33{,}000}{V \times 101 \cdot 33} = 189\,\frac{P}{V}.$$

Let it be also assumed that the H.P. in the shaft at the propeller = ·82 I.H.P.
Then the athwartship moment of each blade

$$= \frac{\cdot 82\,P \times 33{,}000}{2\,\pi\,R} = 4300\,\frac{P}{R}.$$

Assume that the centre of pressure is at $\frac{2}{3}$ radius $= \frac{D}{3}$.

Then the athwartship thrust per blade

$$= \frac{4300\,\frac{P}{R}}{\frac{D}{3}} = 12{,}900\,\frac{P}{R\,D}.$$

Having obtained these two components of the thrust, the resultant thrust at right-angles to the root of the blade may be determined. By equating the moment of this thrust about the root to the

moment of resistance of the section of the root, the formula for the thickness h becomes

$$h^2 = c\mathrm{P} \cdot \frac{\mathrm{D} - d}{b}\left(\frac{d}{p\mathrm{V}} + \frac{20}{\mathrm{RD}}\right),$$

where c = 2 for steel or manganese bronze;
" = ·5 for gun-metal;
" = 6 for cast iron.

The symbols are the same as in the preceding formula.

The elliptical arc of Fig. 37 at any radius being unrolled into a straight line, the thickness as found is set off at its middle point, and a circular arc described through the point and the extremities of the unrolled ellipse will give the thickness at other points than the middle (see Fig. 41).

When the blades are made separate from the boss, they are usually attached to it by bolts arranged in a bolt circle of diameter C inches. As the thickness of the metal at the root of the blade is not disposed symmetrically about the centre of the flange, it is convenient to put one more bolt on the after side than on the forward side, and as the after bolts take the ahead load, which is greater than the astern, it is also an advantageous arrangement. If B be the combined area in square inches of the bolts at the smallest section (at the bottom of the thread) on the after side of the blade, h the distance of the centre of pressure of the blade from the under side of the

flange in feet, R and P being as before, then B = $\frac{P.h.K}{C.R}$, the coefficient K ranging in value from 18 to 21 for gun-metal.

The number of bolts having been settled, their diameter is at once found. The bolts on the forward side are usually made of the same diameter

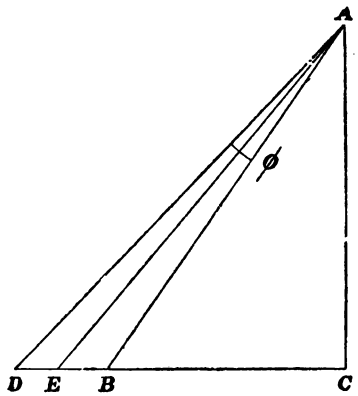

FIG. 49.

for convenience. For the sake of possible adjustments that may be desirable on the trials of the machinery, and also for fining the pitch when it becomes necessary to reduce the steam pressure in the boilers, it is customary to make the bolt-holes in the blade-flange oval in order that the inclination of the blades to the axis may be altered. The amount of the oval is thus determined. Let the true pitch of the blade be 2π C E, Fig. 49, and let the desired range of pitch be 2π E B and 2π E D respectively, on each side of this pitch.

Take a radius C A, so that A is about half-way up the blade. Join A D, A E, A B, then if ϕ be the angle D A. B, the holes in the flange must be so elongated as to admit of the blade being turned through $\frac{\phi}{2}$ on each side. The amount of the elongation on each side is therefore

$$\frac{\pi C}{360} \cdot \frac{\phi}{2} = \pi C \frac{\phi}{720},$$

ϕ being measured in degrees.

If the axis of the helices of intersection of the blade be originally that of the propeller shaft, the acting surface will not be truly helical about the centre line of the shaft when the blade is turned round as just described, since the pitch is not altered uniformly. There are only two sections of the blade which receive the same change of pitch, and these are situated at the radii corresponding to a pitch angle of 45° in the case of the original and modified pitches respectively. Sections between these points receive a less change of pitch, and sections outside them a greater, in proportion to their distance from them.

The effect produced, therefore, by twisting through any given angle depends upon the pitch-ratio; if this is small the critical points are near the boss, and twisting to augment pitch, for example, causes the pitch to increase throughout the greater part of the length of the blade, the maximum occurring at the tip. If the pitch-ratio be such

that the critical points fall about the middle of the length, twisting to fine the pitch will then result in a blade having the maximum pitch in the centre.

Figs. 38 and 40 show respectively longitudinal and transverse sections through the propeller boss.

CHAPTER VII.

THE HYDRAULIC PROPELLER.

THERE are many reasons why a hydraulic propeller would be preferred to a screw or paddle in certain cases, if an economical result could be obtained with it, but the efficiency of the apparatus is necessarily so small that at the present time it is believed that there is not a single vessel using this propeller for warlike or commercial purposes. When economy is a secondary consideration, and the circumstances are such as apparently to preclude the use of any propeller external to the vessel, the hydraulic propeller finds its opportunity.

A steam lifeboat, built by Messrs. R. and H. Green, of Blackwall, in 1888, for the National Lifeboat Institution, was fitted with hydraulic machinery made by Messrs. J. I. Thornycroft and Co., and two other similar vessels have since been built and engined by the latter firm, and have met with a considerable measure of success.

It was considered that the difficulty of keeping a screw immersed, and the danger of its becoming fouled by wreckage, or injured upon a sandbank,

rendered it unsuitable, and justified the use of a propeller which could never race, and which was much less liable to injury.

The term jet-propeller, although in common use, is incorrect, the propeller in this system of propulsion being a pump within the vessel, which discharges jets of water in a sternward direction, which are analogous to the race of a screw or paddle.

In 1879, a hydraulic vessel called the *Hydromotor*, was built in Germany from the designs of Dr. Fleischer.

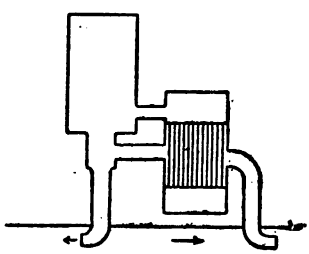

Fig. 50.

In this vessel the engine and pump were combined, the arrangement being as follows :—

There was a cylinder, lined inside with wood, at the bottom of which was a large pipe leading to a nozzle at the bottom of the vessel. A float of nearly the same diameter as the cylinder worked up and down in it. The cylinder being full of water, and the float consequently at the top, steam was admitted by a valve above the float, and, driving it down, ejected the water through the nozzle, Fig. 50.

On reaching the bottom of its stroke, the float opened the exhaust, and the steam passed into the condenser. The vacuum then created in the

cylinder caused the water to rise partly through the nozzle, but principally through a suction valve in the bottom of the condenser. The cylinder was thus filled with water and the float rose to the top, in doing which it closed the exhaust and opened the steam-valve, and the operation was repeated. The loss by condensation appears to have been less than might have been expected in a cylinder filled alternately with steam and water, but as the

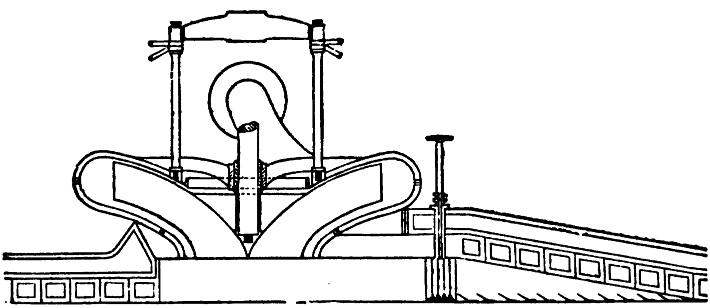

FIG. 51.

cylinder was not entirely emptied at each stroke a layer of boiling water always remained at the top and warmed the wooden lining as the float descended. The information published as to the results obtained was unreliable, as no proper measured mile trials were made. All calculations based upon indicated horse-power cards are of little value in this system, as the loss between the boiler and the indicator, which must be very large, is thereby ignored. The only correct basis for comparison with either a screw or a turbine would be

the amount of steam consumed per unit of work done.

In 1866, two armoured gunboats, the *Viper* and *Waterwitch*, were built for the British

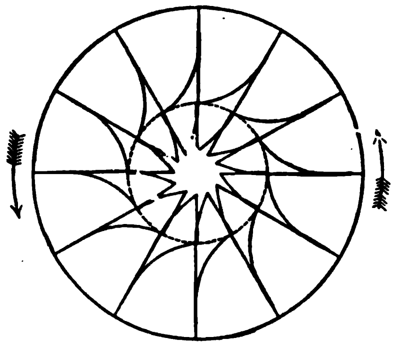

FIG. 52.

Government, the former being propelled by twin-screws and the latter by hydraulic machinery,

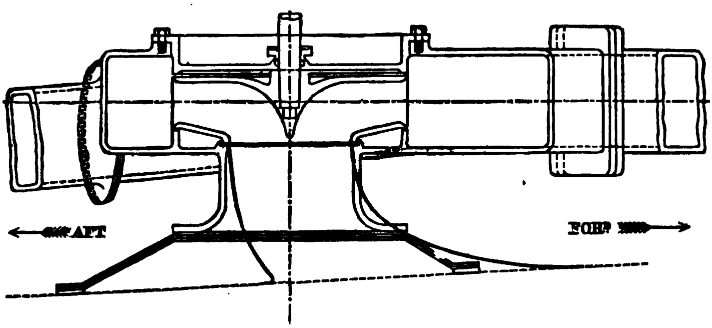

FIG. 53.

designed by Mr. Ruthven. The *Waterwitch's* propeller consisted of a turbine fourteen feet in diameter, see Fig. 52, which drew water in at the bottom of the vessel, and discharged it

through two 24-inch nozzles at the sides level with the water (see Fig. 51).

In 1878 a hydraulic vessel was designed by Captain Lilliehöök, of the Swedish Navy, for

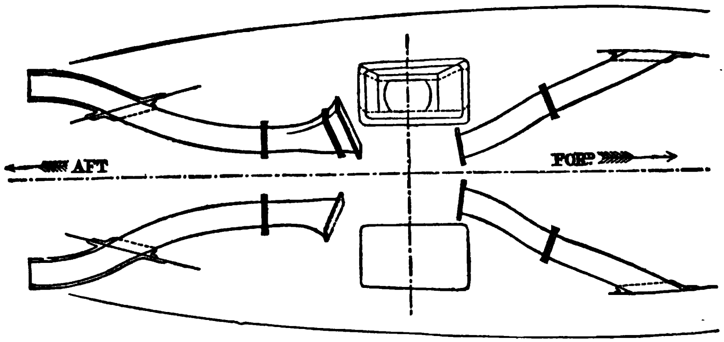

FIG. 54.

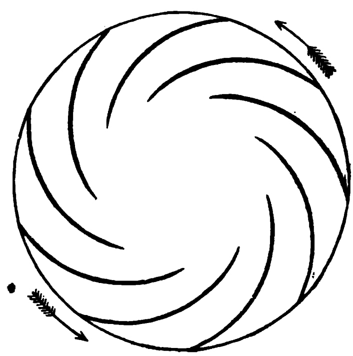

FIG. 55.

competition with a similar vessel with twin-screws. The hydraulic vessel was propelled by two turbines about two feet in diameter, which discharged water through submerged orifices at the sides near the extremities (see Figs. 53, 54, and 55).

In 1883, Messrs. J. I. Thornycroft and Co. fitted one of a number of second-class torpedo boats they

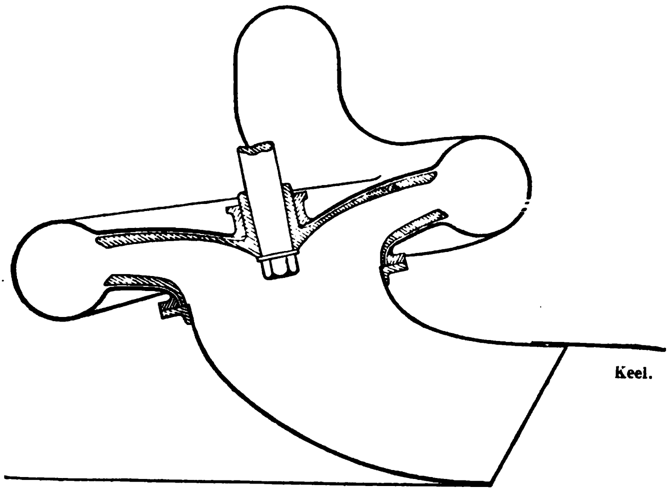

FIG. 56.

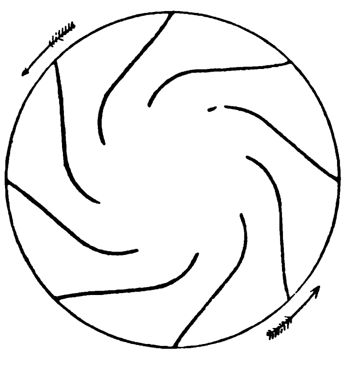

FIG. 57.

were building for the British Navy with a hydraulic propeller consisting of a turbine 2 feet 6 inches in diameter, which discharged through

two 9-inch nozzles at the sides above water (see Figs. 56 and 57 and Plate V).

All these vessels were fully described by the author in a paper read before the Institution of Civil Engineers,* from which the figures and Tables V. and VI., page 172, giving a detailed comparison of the performance of the respective screw and turbine vessels, are taken. In every case a very remarkable difference was found between them, and always to the disadvantage of the latter. There appears to be a loss of power corresponding to about 50 per cent. experienced by the hydraulic propeller as compared with the screw. The causes of this loss are not hard to find.

In both the *Waterwitch* and the Swedish boat, the water was received into the ship through a hole in the bottom in such a way as to suddenly arrest all the velocity which it had relatively to the vessel.

In other words, the entering water struck the after side of the inlet, and had the velocity of the ship impressed upon it before it entered the turbine. If the inlet is formed in the shape of a scoop, as was done in the Thornycroft boat (see Fig. 56 and Plate V.), and the water caused to change its direction gradually, without having its velocity relative to the ship destroyed, then this cause of loss is avoided. In such a case if the vessel were towed along with the turbine removed,

* Proc. Inst. Civil Engineers, vol. lxxvii. p. 1.

and replaced by a curved channel connecting the inlet and outlets, the water would be scooped up, and would flow out at the nozzles, leaving them, if they are not above the surface and if the friction of the passages could be eliminated, with a velocity relative to the ship equal to the speed of the ship and with no velocity relative to still water.

The inlet of the Swedish vessel was subsequently altered, as shown in thick black lines in Fig. 53, and the partial scoop thus formed caused the speed to be increased from 7·87 knots to 8·12 knots, with the same expenditure of power.

Another loss of efficiency in the hydraulic system is due to the small area of stream acted upon, and the consequently high velocity which has to be imparted to it in order to give the necessary reaction (see page 4). The reason why the area of stream acted upon is necessarily small, is that the size of the orifice which it is possible to make in a ship's bottom is restricted by structural considerations, and must be very small indeed compared to the area of a screw's disc. Then again the weight of water admitted into the ship is a serious consideration, as it represents so much loss of displacement.

A further waste of power is caused by the friction of the water in the pipes and passages, and by the changes in direction of its flow in passing through the bottom and out through the

sides in a fore and aft direction. For these reasons the hydraulic propeller is essentially wasteful.

In the screw and turbine competitive Thornycroft torpedo boats, the efficiencies were found to be as follows. Screw-boat: engine, 0·77; screw-propeller, 0·65; total efficiency, 0·5. Hydraulic boat: engine, 0·77; pump, 0·46; jet, 0·71; total efficiency, 0·254.

The efficiencies of the pump and jet in this boat were measured by the author in the following manner. A thin plate, $1\frac{5}{16}$ inch square, was attached to the end of a thin lever and placed in the jet just where it left the nozzle. The pressure on this plate was recorded by a dynamometer attached to the end of the lever. By finding the pressure upon a similar lever without the plate, the effect of the portion of the lever immersed in the jet could be allowed for.

The apparatus was so arranged that the pressure could be measured not only in the centre but at a number of different positions in the jet. From the pressures on the plate the velocity of the stream at different parts of the jet was estimated, and from the mean velocity, the quantity of water discharged was calculated

The relation between velocity of jet and pressure on the plate was taken to be as follows:—

$$\text{Pressure on plate} = \frac{0\cdot627 \times (\text{area}) \times \text{heaviness of fluid} \times (\text{velocity})^2}{\text{gravity}}.$$

If W = weight of water discharged per second;
V = speed of vessel in feet per second;
S = true slip or acceleration, or additional velocity impressed by the propelling apparatus in feet per second;
V + S = velocity of discharge in feet per second.
g = acceleration produced by gravity in feet per second = 32·2;

Then the theoretical efficiency of the jet $= \dfrac{V}{V + \dfrac{S}{2}}$.

The efficiency of pump, supposing jet to have the theoretical efficiency and the engine an efficiency of 0·77, is

$$= \frac{\text{Work stored up in water}}{\text{Effective H.P. of engine}} = \frac{\dfrac{W\,V\,S}{g} + \dfrac{W\,S^2}{2\,g}}{\text{I.H.P.} \times 550 \times 0 \cdot 77}.$$

The efficiency of pump and jet

$$= \frac{\text{Useful work in jet}}{\text{Effective H.P. of engine}} = \frac{\dfrac{W\,V\,S}{g}}{\text{I.H.P.} \times 550 \times 0 \cdot 77}.$$

The total efficiency

$$= \frac{\text{Useful work in jet}}{\text{Work expended}} = \frac{\dfrac{W\,V\,S}{g}}{\text{I.H.P.} \times 550}.$$

TABLE V.

—	Date.	Length.	Beam.	Maximum Draught.	Displacement.	I.H.P.	Speed per Hour.	Midship Section.	$\frac{V^3 \times D^{\frac{2}{3}}}{I.H.P.}$	Number of Propellers.	Revolutions per minute.
		ft. in.	ft. in.	ft. in.	tons.		knots.	sq. ft.			
H.M.S. *Viper*	1867	162 0	32 0	11 10	1,180·00	696	9·58	337·0	141·4	2 screws	110
H.M.S. *Waterwitch* ..	1867	162 0	32 0	11 2	1,161·00	760	9·30	336·0	116·9	1 turbine	40
Swedish screw ..	1878	58 0	10 9	4 3	20·00	90	10·00	25·0	82·0	2 screws	250
„ hydraulic ..	1878	58 0	10 9	4 2	21·00	78	8·12	25·0	52·5	2 turbines	384
Thornycroft screw ..	1883	63 0	7 6	3 8½	12·89	170	17·30	11·9	169·0	1 screw	636
„ hydraulic	1883	66 4	7 6	2 6	14·40	167	12·60	13·4	72·0	1 turbine	428

TABLE VI.

—	Diameter of Turbine.	Area of Inlet.	Combined Area of Discharge.	$\frac{\text{Midship Section.}}{\text{Area of Discharge.}}$	Velocity of Discharge.	Water Discharged.	Efficiency of Pump and Jet.	Total Efficiency.
	ft. in.	sq. ft.	sq. ft.	sq. ft.	ft. per sec.	lbs. per sec.		
H.M.S. *Waterwitch*	14 0	28·25	6·280	53·5	29·0	11,650	0·234	0·180
Swedish hydraulic	1 11½	1·62	0·864	30·5	28·0	1,510	0·277	0·214
Thornycroft hydraulic ..	2 6	1·52	0·951	14·1	37·2	2,210	0·330	0·254

CHAPTER VIII.

THE SCREW-TURBINE PROPELLER.

THIS propeller was the fruit of the study given to the subject of hydraulic propulsion by Mr. Thornycroft, when designing the hydraulic torpedo boat already mentioned.

It has been pointed out in the preceding chapter that there are four characteristics of the centrifugal pump, as applied for the purpose of propelling vessels, which prevent it from competing successfully with the paddle or the screw. These are:—

1. The difficulty of getting the water through the bottom of the vessel and into the pump without checking the velocity which it already has relative to the vessel.

2. The necessity of carrying in the vessel all the water acted upon.

3. The loss caused by friction of the water in the pipes.

4. The loss due to bends in the passages.

It was obvious that if the turbine could be put outside the vessel and under the bottom, the first two causes of loss would be avoided, and also that, if the water could be made to flow axially through

the turbine instead of radially, as in the Ruthven pump, no pipes would be needed, there would be no loss by changes of direction, and the friction would be reduced to that due to passage through the turbine alone.

The propeller illustrated in Fig. 58 was therefore devised upon these lines, and as it is neither a screw nor a turbine strictly speaking, although allied to both, it has received the name of the screw-turbine.

It consists of a cylinder containing within it a body or boss of such a shape that the channel is gradually contracted from the forward to the after end. Within the forward part of the cylinder there are revolving screw-blades attached to the forward part of the boss, which is keyed on to the shaft, and is separate from the after part. The pitch of the forward edge of the screw-blades multiplied by the number of revolutions is approximately equal to the velocity of feed; the pitch increases uniformly along the length of the blade, imparting a uniform acceleration to the water. Aft of the revolving blades are numerous guide-blades of contrary curvature, which are fixed to the rearward portion of the boss and to the cylinder. The cylinder is attached to the stern of the vessel. The area of the channel through the propeller is so proportioned as to suit the acceleration of the water caused by the blades. Thus, at the forward end is a large opening which will admit a certain

quantity of water at the velocity of feed, at the after end the area is restricted to that necessary to

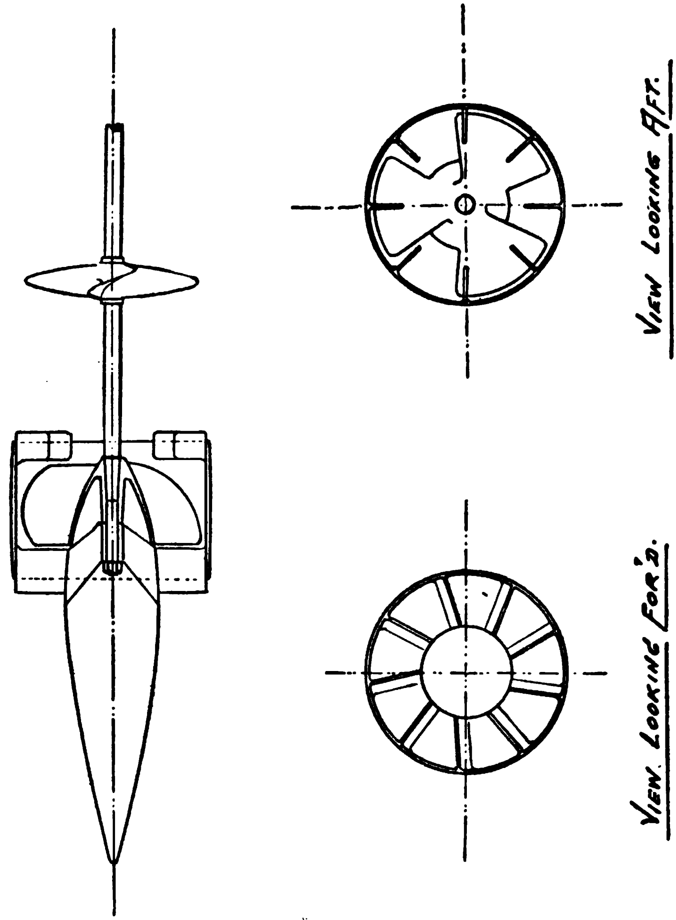

Fig. 58.

allow of the exit of this water at the speed of discharge.

The long tapering body, forming a prolonga-

tion of the boss outside of the cylinder, allows the annular stream of water to close gradually without the formation of eddies. As the long pitch of the screw blades causes considerable rotation of the water, the guides are so formed as to direct the water into a straight line aft, and the rotary motion is utilised without loss, except that caused by friction on the surface of the guides. The thrust delivered by the curved guides amounts to about one-third of the whole thrust.

Forward of the cylinder, and keyed on to the shaft, are screw blades of the same radius as the cylinder, whose function is to propel when going astern.

They are of the same pitch as the leading edge of the turbine propeller blades, and advance through the water without propelling when the vessel is going ahead. When the vessel goes astern, these blades receive water not only through the turbine cylinder, but also from outside it. The speed of any vessel for a given number of revolutions is always much less when going astern than when going ahead, so that, when the engines are reversed, the blades throw a stream of water forward and thus propel the vessel astern.

If we compare the screw-turbine with the propeller shown in Fig. 7, it will be seen that instead of occupying, as that does, a very short length of the contracting column, the whole contraction is forced to take place within the length of the

cylinder. The column enters it at the speed v and leaves it at the speed $v + s$.

The efficiency, therefore, neglecting friction, is equal to that of the ideal screw $\frac{v}{v + \frac{s}{2}}$. But as the surfaces are large the efficiency is reduced by friction to rather less than that of the best ordinary screw, over which, however, it has this advantage, that, as there is no suction in front of the cylinder, there is no augmentation of hull resistance.

The chief value of the propeller lies in the fact that its maximum efficiency is obtained with a very high slip-ratio. Thus, while the thrust at maximum efficiency of an ordinary three-bladed screw of one foot diameter as ascertained by model experiments was 12 lbs., the thrust of a screw turbine of the same diameter and equal efficiency, was 26½ lbs., while that of another of the same diameter, but of longer pitch and less efficiency, was as high as 32 lbs. The same thrust, and nearly the same efficiency, can therefore be obtained with a screw-turbine as with an ordinary screw of much larger diameter, and it is therefore especially adapted to vessels of light draught.

The arrangement shown in Plate VI. has proved very successful when the draught of water is small.

The launch illustrated has a draught of 12 inches.

A tunnel is formed in the bottom of the boat, the top of which rises above the surface, the ends being submerged. A 16-inch screw-turbine is placed in the tunnel so that one-fourth of the diameter of the propeller is above the water-level when the boat is at rest, but, as soon as it moves, water is drawn up into the tunnel and the air expelled by the action of the propeller, which then works completely submerged.

There is no loss of power in lifting the water four inches above the level of the surface, because in falling it gives out the work expended in raising it. There is an incidental convenience in this arrangement. An air-tight door can be placed at the crown of the tunnel, immediately over the propeller, which can be opened from inside the boat, since the admission of air to the tunnel causes the water within it to fall to the level of the outside water surface and leaves the propeller partially emerged. It can then be examined and cleared if it should have become fouled, and if there are twin screws this operation can be performed upon one propeller while the other is revolving slowly.

Some vessels, 140 feet by 21 feet, and having a draught of water of 2 feet only, have been built upon this plan by Messrs. Thornycroft. They were propelled by twin-screw turbines 32 inches

diameter, and attained a speed of $15\frac{1}{4}$ knots. If ordinary screws had been employed they would have required to be 54 inches in diameter to have run at the same number of revolutions per minute,

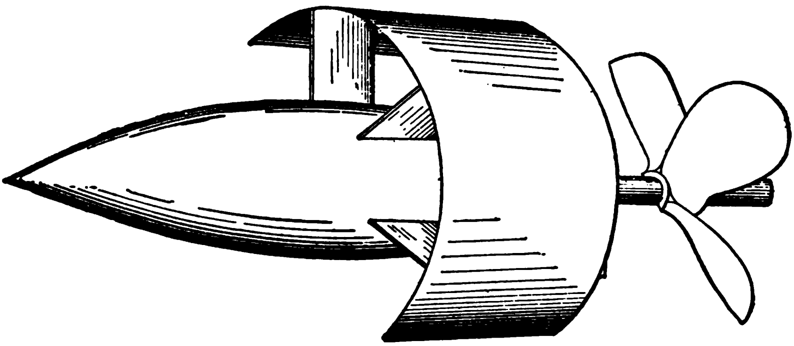

FIG. 59.

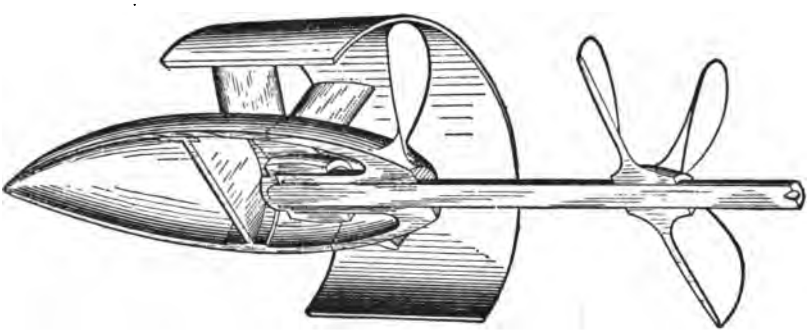

FIG. 60.

and given the same thrust. The efficiency would have been a little greater, as the screw-turbines were worked at rather more than their proper slip-ratio.

A launch, 56 feet long and 15 inches draught

of water, has attained a speed of 16¼ knots with one screw-turbine 20 inches in diameter.

Figs. 59 and 60 show the screw-turbine in section.

Fig. 61 shows an arrangement of the screw-turbine suitable for some purposes, such, for example, as ferry steamers of shallow draught. The stern-way screw, being placed in the bow, is in the most favourable position for propelling and steering astern. It has been found that when vessels have had screws at each end, both of which took part in

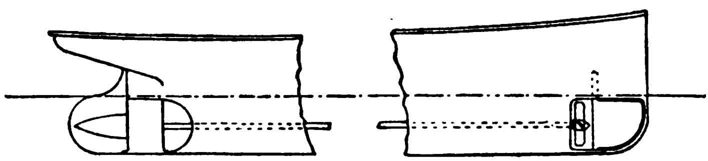

FIG. 61.

propelling ahead, the resistance has been largely increased by the action of the forward screw, the augment of resistance amounting in some cases to about 23 per cent.; but with a screw-turbine at the stern doing the whole work going ahead, and a stern-way screw at the bow advancing without slip, no water would be thrown against the vessel, and the resistance would not be increased.

In ferry steamers, which by the nature of the service are compelled to make the passage in one direction ahead, and in the other astern, and therefore should be able to travel equally well either

way, the best arrangement appears to be to place two ordinary screws at each end, making the vessels double-ended. The augmented resistance due to the two forward screws is less than that caused by a single screw in the forefoot.

Table VII.—Values of V^5.

V	V^5	V	V^5	V	V^5	V	V^5
1·0	1·000	5·6	5,507	10·2	110,410	14·8	710,080
·1	1·610	·7	6,017	·3	115,930	·9	734,400
·2	2·488	·8	6,563	·4	121,670	15·0	759,370
·3	3·713	·9	7,149	·5	127,630	·1	785,030
·4	5·378	6·0	7,776	·6	133,820	·2	811,370
·5	7·594	·1	8,446	·7	140,260	·3	838,410
·6	10·49	·2	9,161	·8	146,930	·4	866,170
·7	14·20	·3	9,924	·9	153,860	·5	894,660
·8	18·90	·4	10,737	11·0	161,050	·6	923,900
·9	24·76	·5	11,603	·1	168,520	·7	953,890
2·0	32·00	·6	12,523	·2	176,230	·8	984,660
·1	40·84	·7	13,501	·3	184,250	·9	1,016,200
·2	51·54	·8	14,539	·4	192,540	16·0	1,048,600
·3	64·36	·9	15,640	·5	201,140	·1	1,081,700
·4	79·63	7·0	16,807	·6	210,030	·2	1,115,800
·5	97·66	·1	18,042	·7	219,250	·3	1,150,600
·6	118·8	·2	19,349	·8	228,780	·4	1,186,400
·7	143·5	·3	20,731	·9	238,640	·5	1,223,000
·8	172·1	·4	22,190	12·0	248,830	·6	1,260,500
·9	205·1	·5	23,731	·1	259,370	·7	1,298,900
3·0	243·0	·6	25,355	·2	270,270	·8	1,338,300
·1	286·3	·7	27,068	·3	281,530	·9	1,378,600
·2	335·5	·8	28,872	·4	293,500	17·0	1,419,800
·3	391·4	·9	30,771	·5	305,140	·1	1,462,100
·4	454·3	8·0	32,768	·6	317,580	·2	1,505,400
·5	525·2	·1	34,867	·7	330,880	·3	1,549,600
·6	604·7	·2	37,074	·8	343,600	·4	1,594,900
·7	693·5	·3	39,391	·9	357,230	·5	1,643,200
·8	792·4	·4	41,822	13·0	371,290	·6	1,688,700
·9	902·2	·5	44,370	·1	385,790	·7	1,737,300
4·0	1,024	·6	47,042	·2	400,740	·8	1,786,900
·1	1,159	·7	49,843	·3	416,150	·9	1,837,700
·2	1,307	·8	52,773	·4	432,040	18·0	1,889,600
·3	1,470	·9	55,841	·5	448,400	·1	1,942,600
·4	1,649	9·0	59,049	·6	465,260	·2	1,996,900
·5	1,845	·1	62,403	·7	482,620	·3	2,052,400
·6	2,060	·2	65,908	·8	501,070	·4	2,109,100
·7	2,293	·3	69,568	·9	518,890	·5	2,167,000
·8	2,548	·4	73,391	14·0	537,820	·6	2,226,200
·9	2,825	·5	77,378	·1	557,310	·7	2,286,700
5·0	3,125	·6	81,537	·2	577,350	·8	2,348,500
·1	3,450	·7	85,874	·3	597,970	·9	2,411,600
·2	3,802	·8	90,390	·4	619,170	19·0	2,476,100
·3	4,182	·9	95,099	·5	640,970	·1	2,542,000
·4	4,592	10·0	100,000	·6	663,380	·2	2,609,200
·5	5,033	·1	105,100	·7	686,410	·3	2,677,800

TABLE VII.—VALUES OF V^5—*continued.*

V	V^5	V	V^5	V	V^5
19·4	2,748,000	24·0	7,962,600	28·6	19,135,000
·5	2,819,500	·1	8,129,900	·7	19,472,000
·6	2,892,500	·2	8,300,000	·8	19,814,000
·7	2,967,100	·3	8,472,900	·9	20,160,000
·8	3,043,200	·4	8,648,700	29·0	20,511,000
·9	3,120,800	·5	8,827,300	·1	20,867,000
20·0	3,200,000	·6	9,008,800	·2	21,228,000
·1	3,280,800	·7	9,193,600	·3	21,594,000
·2	3,363,200	·8	9,381,200	·4	21,965,000
·3	3,447,300	·9	9,571,900	·5	22,341,000
·4	3,533,100	25·0	9,763,600	·6	22,723,000
·5	3,620,500	·1	9,962,500	·7	23,109,000
·6	3,709,700	·2	10,163,000	·8	23,501,000
·7	3,800,600	·3	10,366,000	·9	23,898,000
·8	3,893,300	·4	10,572,000	30·0	24,300,000
·9	3,987,800	·5	10,782,000	·1	24,708,000
21·0	4,084,100	·6	10,995,000	·2	25,121,000
·1	4,182,300	·7	11,211,000	·3	25,540,000
·2	4,282,300	·8	11,431,000	·4	25,964,000
·3	4,384,300	·9	11,655,000	·5	26,394,000
·4	4,488,200	26·0	11,881,000	·6	26,829,000
·5	4,594,000	·1	12,112,000	·7	27,270,000
·6	4,701,900	·2	12,345,000	·8	27,717,000
·7	4,811,700	·3	12,583,000	·9	28,170,000
·8	4,923,600	·4	12,824,000	31·0	28,629,000
·9	5,037,600	·5	13,069,000	·1	29,094,000
22·0	5,153,600	·6	13,317,000	·2	29,565,000
·1	5,271,800	·7	13,569,000	·3	30,041,000
·2	5,392,200	·8	13,825,000	·4	30.524,000
·3	5,514,700	·9	14,085,000	·5	31,014,000
·4	5,639,500	27·0	14,349,000	·6	31,509,000
·5	5,766,400	·1	14,617,000	·7	32,011,000
·6	5,895,800	·2	14,888,000	·8	32,519,000
·7	6,027,400	·3	15,164,000	·9	33,033,000
·8	6,161,300	·4	15,444,000	32·0	33,555,000
·9	6,297,600	·5	15,728,000	·1	34,082,000
23·0	6,436,300	·6	16,016,000	·2	34,616,000
·1	6,577,500	·7	16,308,000	·3	35,157,000
·2	6,721,100	·8	16,604,000	·4	35,705,000
·3	6,867,200	·9	16,905,000	·5	36,259,000
·4	7,015,800	28.0	17,210,000	·6	36,320,000
·5	7,167,000	·1	17,520,000	·7	37,389,000
·6	7,320,800	·2	17,834,000	·8	37,964,000
·7	7,477,200	·3	18,152,000	·9	38,546,000
·8	7,636,300	·4	18,475,000	33·0	39,135,000
·9	7,798,100	·5	18,803,000	·1	39,732,000

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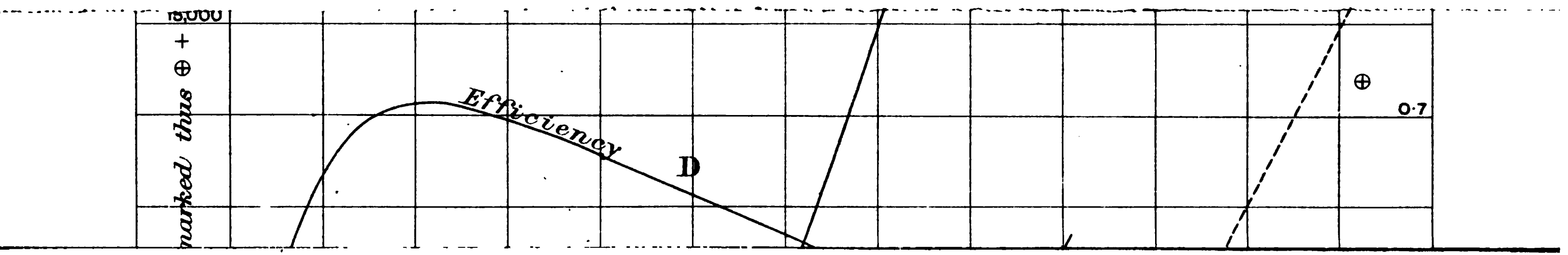
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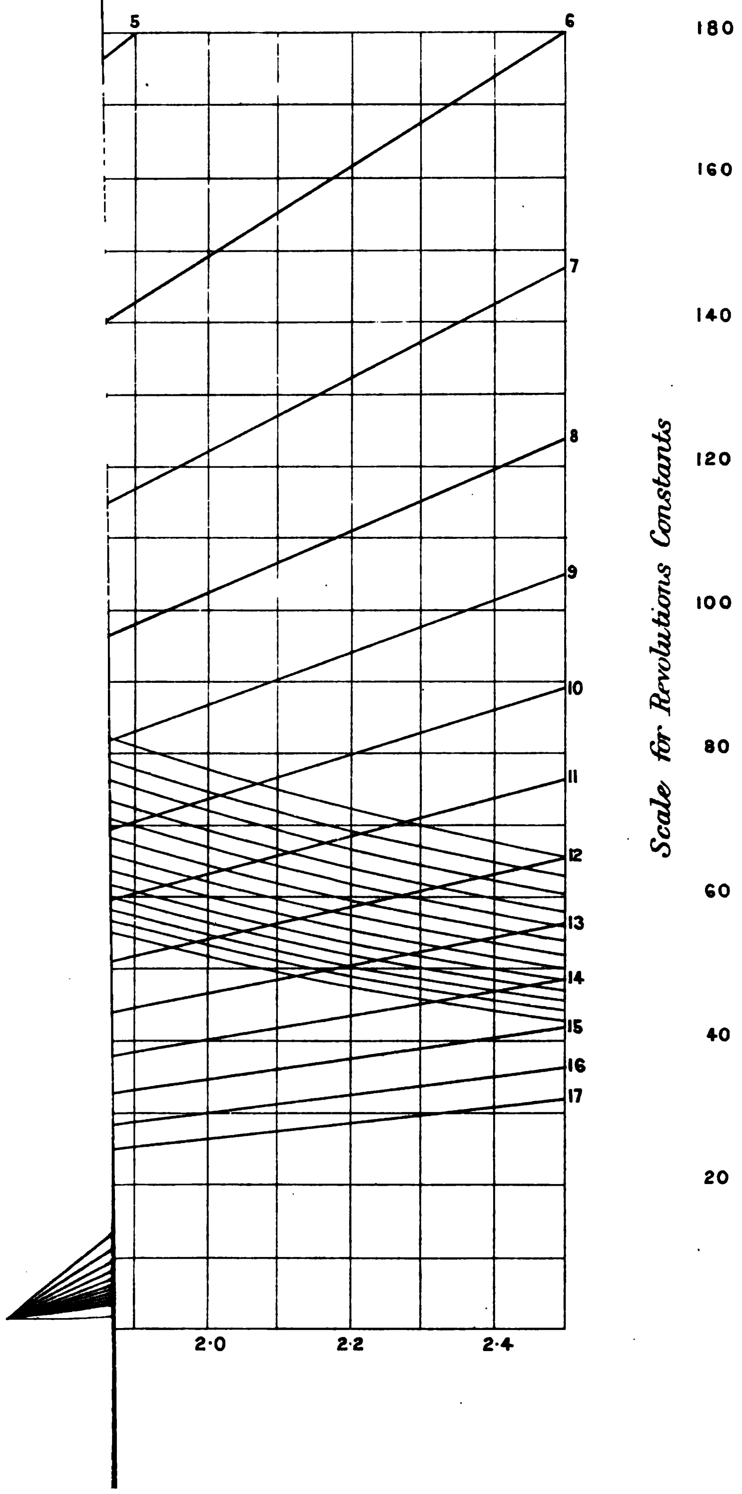
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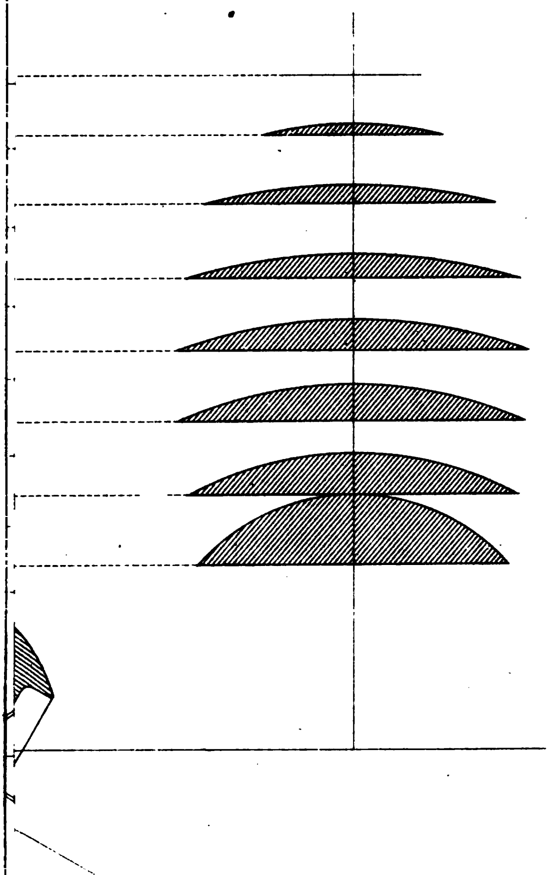

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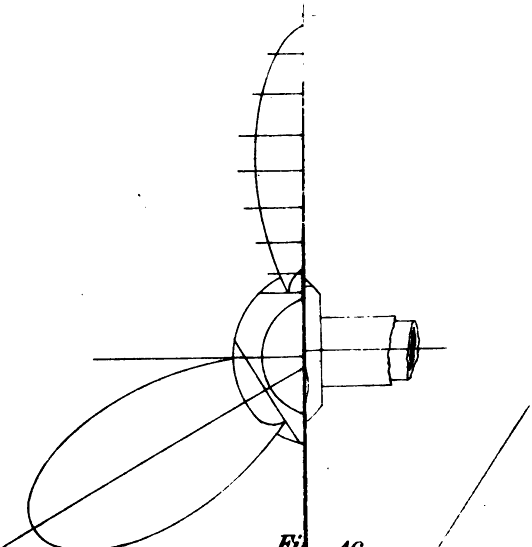

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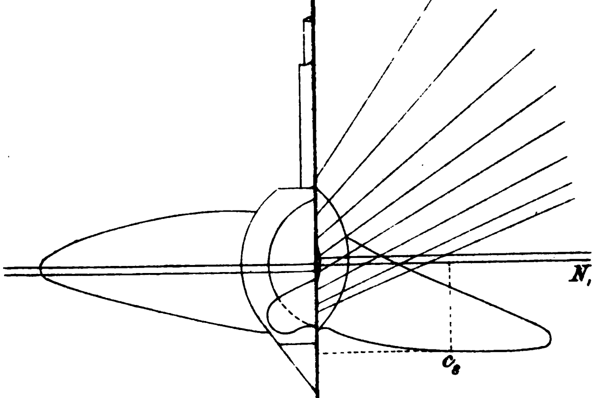

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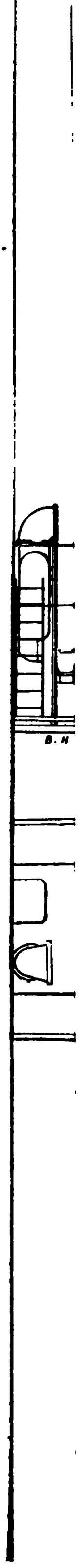
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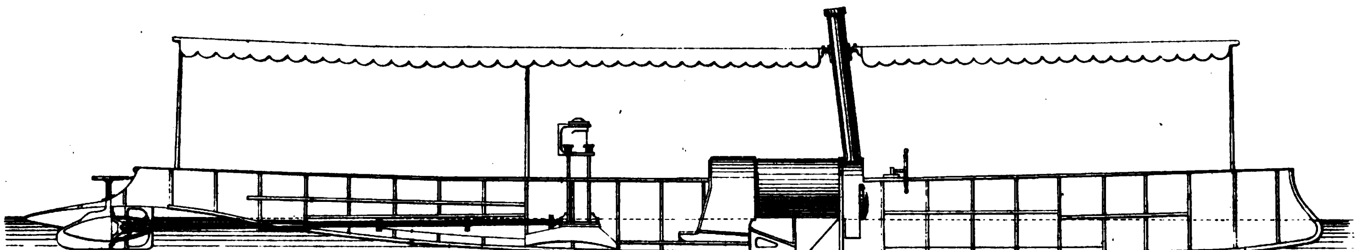
Beam....... 6'. 0"
Draft....... 1'. 0
Speed...... 13 *Knots*



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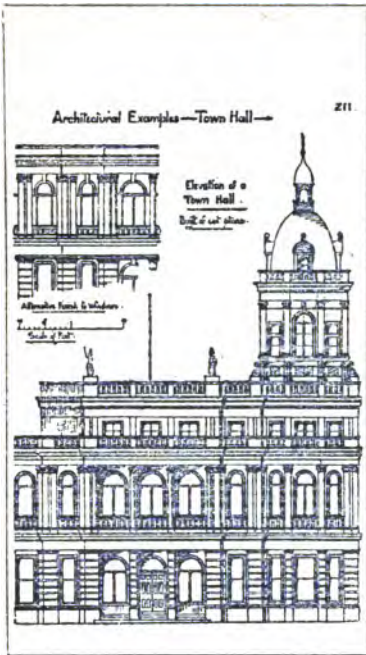

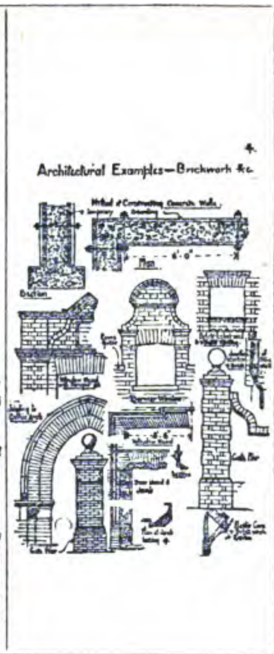

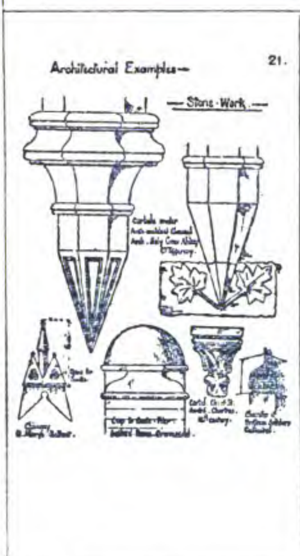

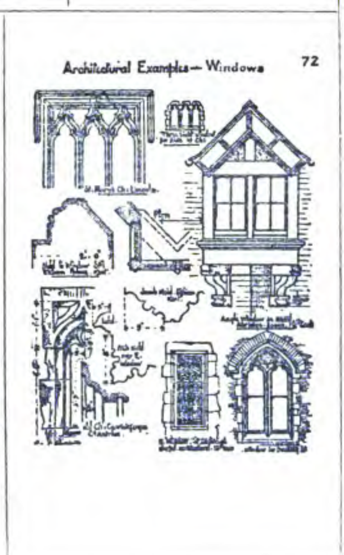

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